AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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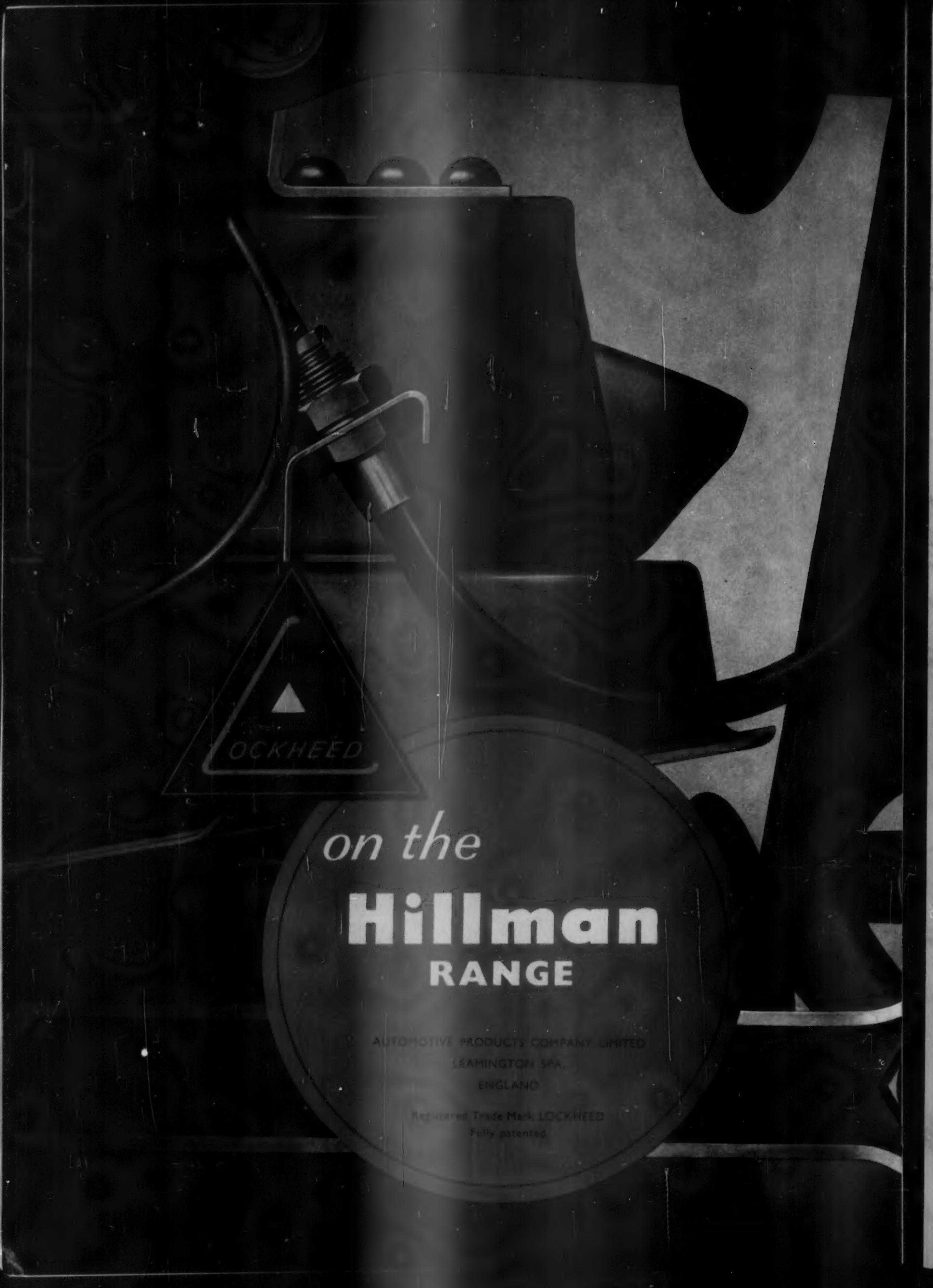

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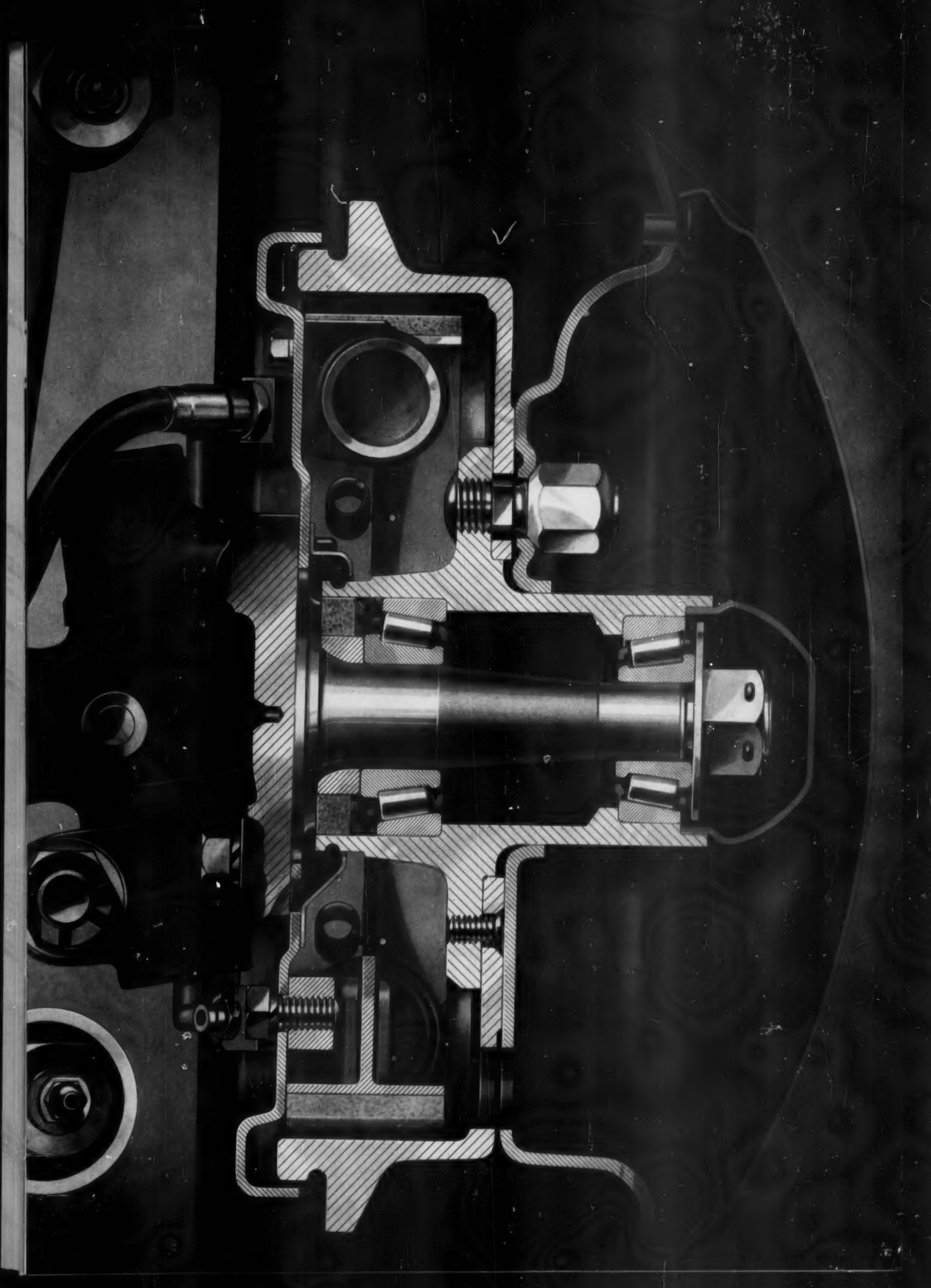
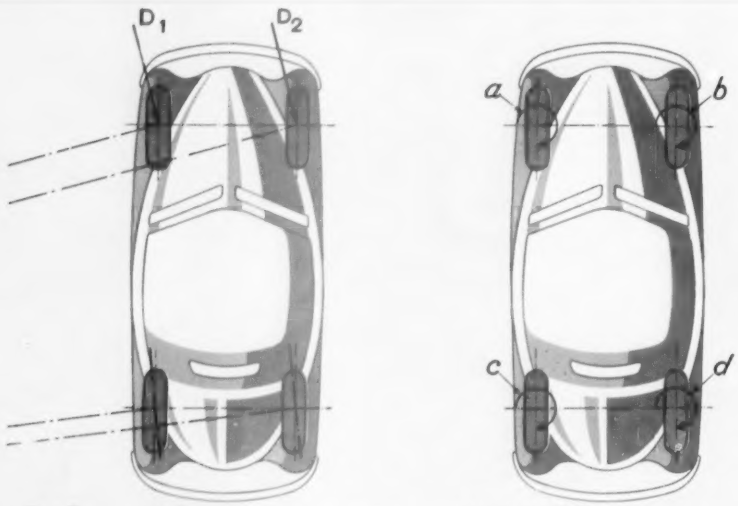


SUBTLETIES OF STEERING

Causes of over- and under-steer: 7. Sideways loads (a)



$\frac{D_2 - D_1}{2} \times \frac{\text{track}}{\text{wheelbase}}$ = extra load to be taken by front wheels due to front wheel drag. The rears provide their contribution, too.

$\frac{a + b + c + d}{\text{wheelbase}}$ = extra sideways load on the front wheels.

At first glance we are tempted to assume that the sideways load carried by a pair of wheels, front or rear, is just the weight carried by those wheels multiplied by the sideways acceleration expressed as a proportion of *g*. Unfortunately it is not quite so simple as this. The centrifugal component of the sideways load carried by a pair of wheels is the major component but the modifications due to other causes are important and cannot be disregarded.

The first of the modifiers is camber thrust. This we have already studied and found to be approximately $W \tan \theta$ where θ is the angle of lean of the wheel in question in relation to the ground, and W is the load carried by that wheel. It would be a remarkable coincidence if any of the wheels of a car remained vertical while cornering. Even the two wheels on a rigid rear axle will lean over to some extent while cornering, because the outer tyre will deflect more, under the increased load it is carrying, than the inner tyre. The angle of tilt of the complete axle caused in this way can be between 1° and 1½° at 0.5*g* sideways acceleration and the camber thrust due to this may therefore be about 5% of the centrifugal thrust.

We have also seen that the presence of a drift angle on a tyre due to the developing of cornering force results in a considerable increase in its drag (an increase of about $C \tan \theta$, where C is the cornering force and θ the drift angle). Since the outer wheels of a car are more heavily loaded than the inner, there will be a preponderance of drag on the outer wheels; this will result in a couple on the car in plan view, which must be resisted by increased sideways force on the front wheels and reduced sideways force on the rear.

We have seen too that the drifting tyre develops a self-righting torque trying to turn it straight. The sum of these self-righting torques for the four wheels also acts on the car as a whole, and must also be resisted by an extra sideways force at the front wheels and a reduced sideways force at the rear wheels.

The total sideways force to be provided by the front wheels is therefore the centrifugal, plus the camber thrust for the pair of wheels, plus the extras due to drag distribution torque and tyre self-righting torques.

The sideways force to be provided by the pair of rear wheels is the centrifugal, plus the camber thrust for the pair of wheels, minus the sideways loads due to drag distribution and tyre self-righting torques.

The manner in which these sideways loads can be related to mean load carried per tyre and weight transference must next be considered. This, in fact, is the final stage, relating the sideways loads to be carried to the tyre properties for whatever mean load is carried and the amount of weight transference which exists.

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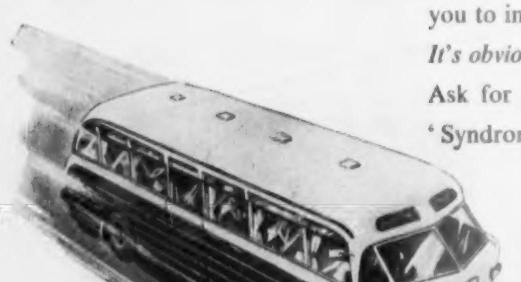

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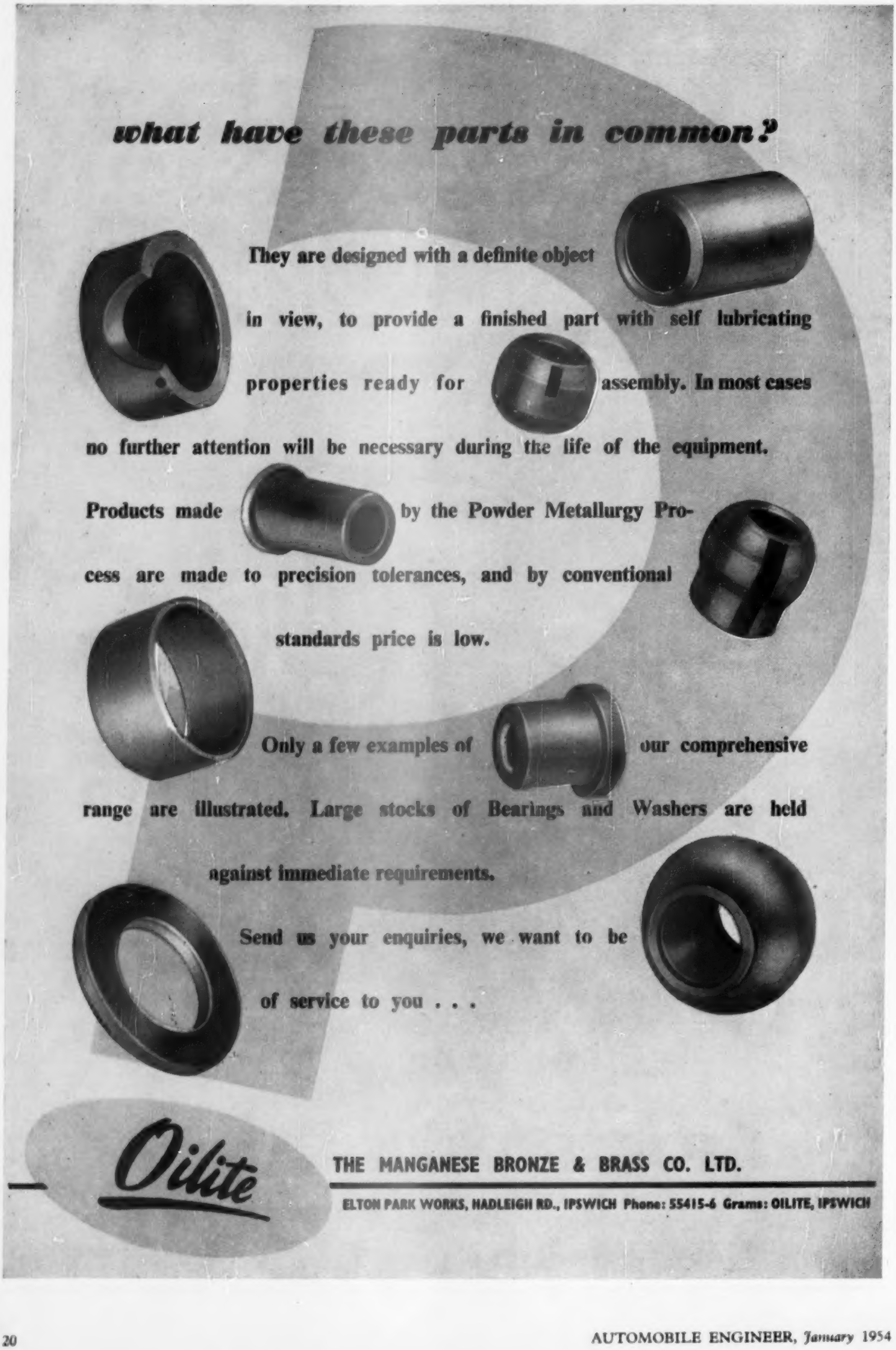
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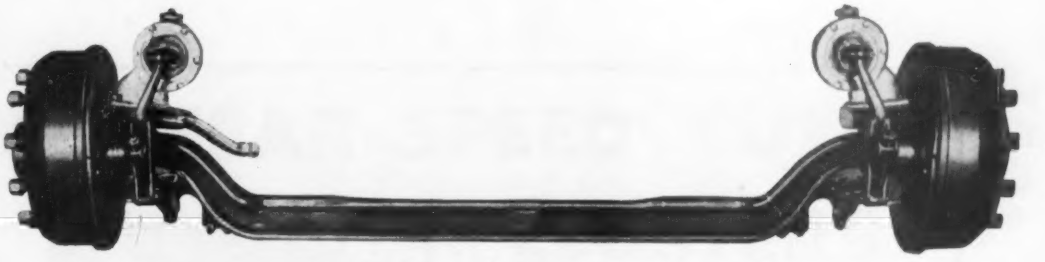

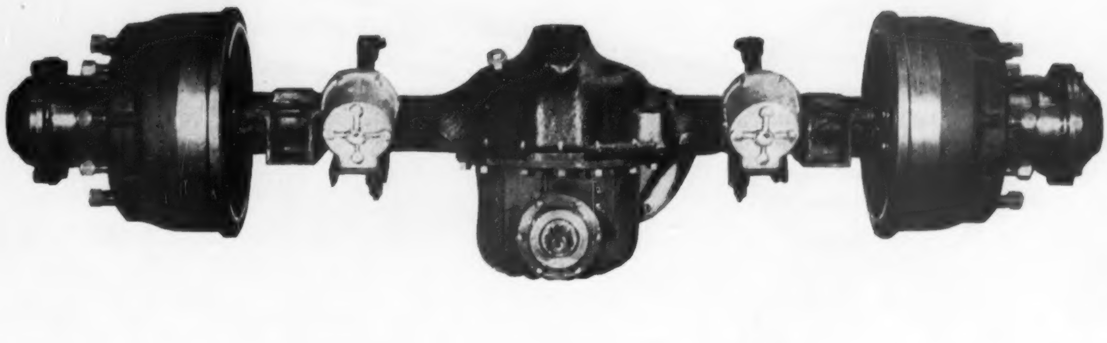


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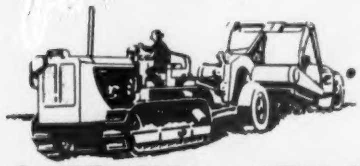

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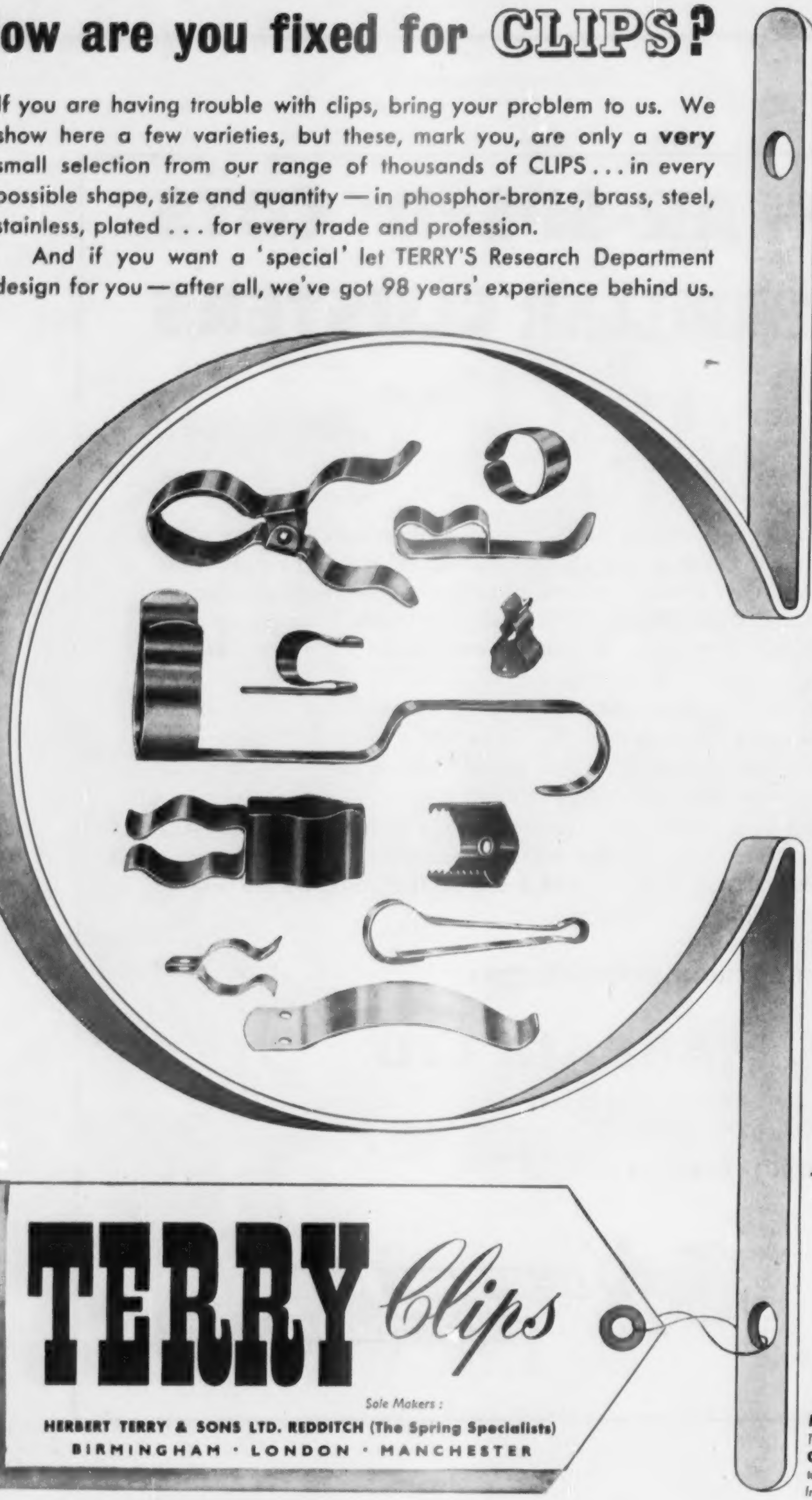

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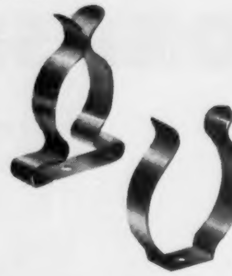

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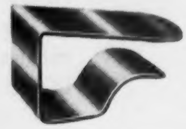

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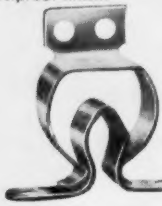

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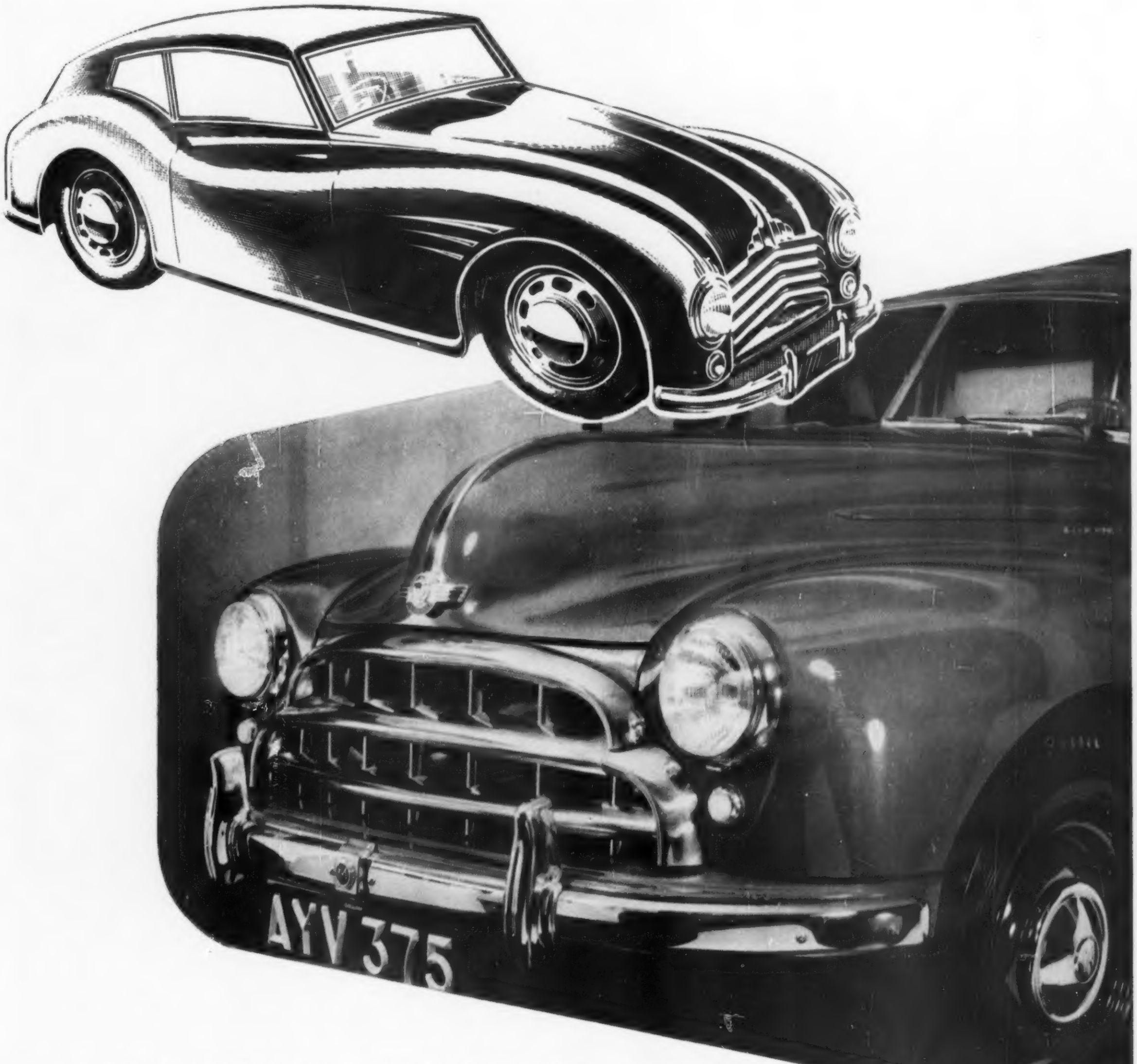
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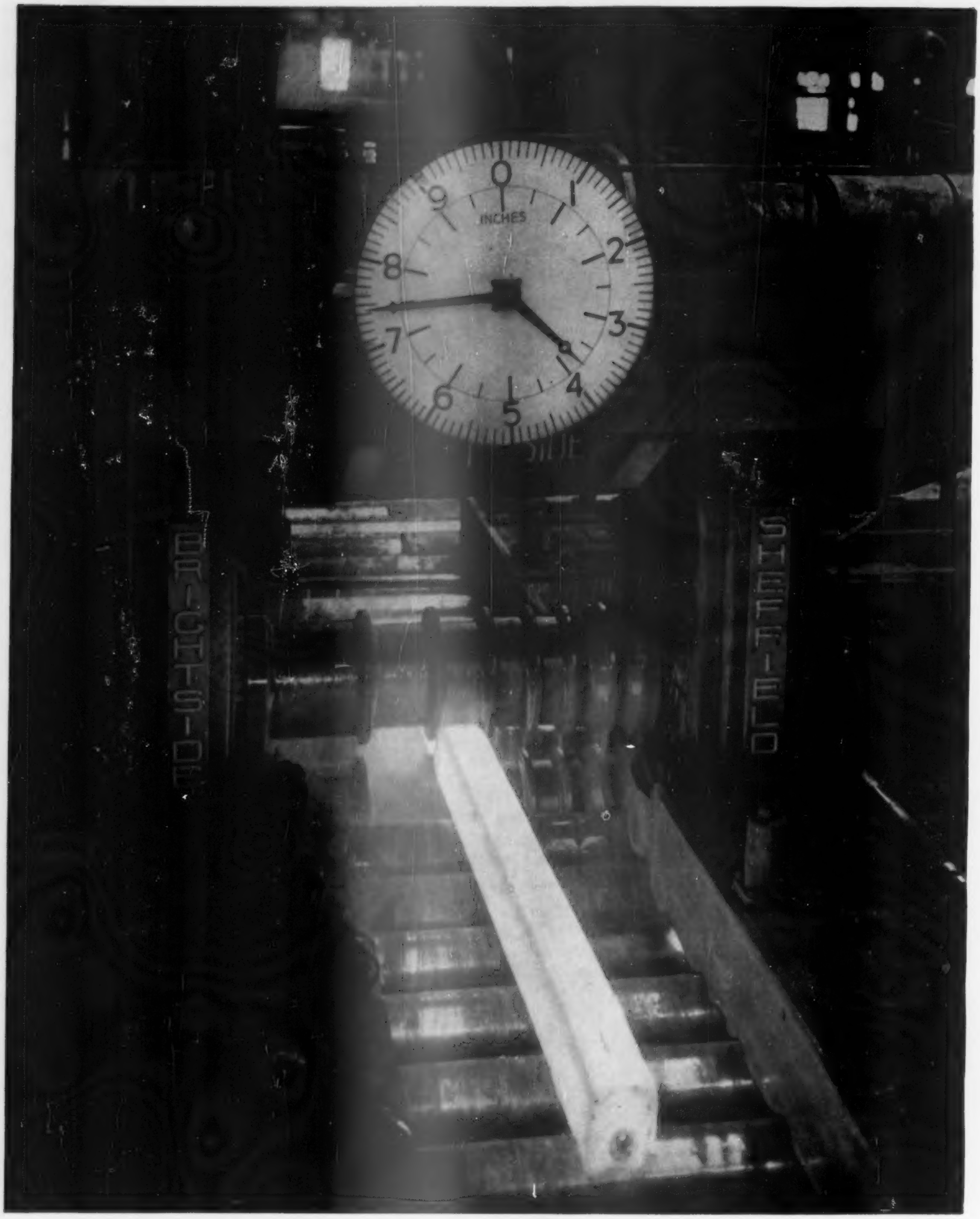

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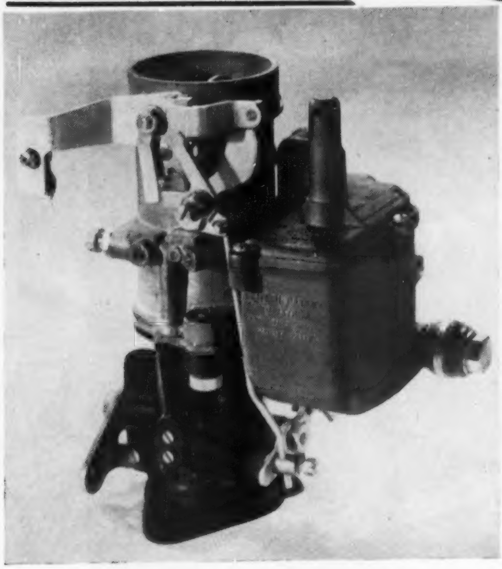

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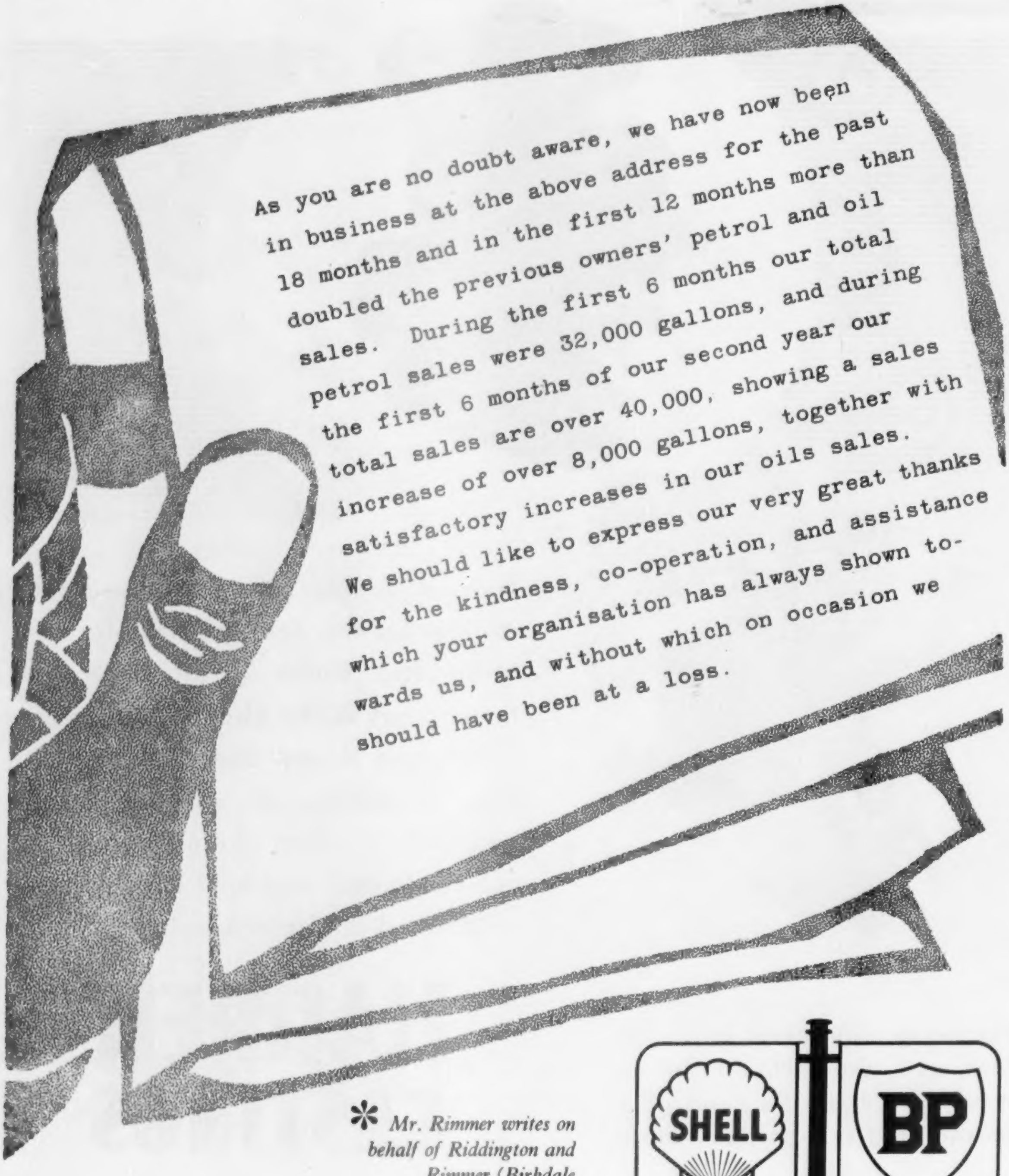

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* *Mr. Rimmer writes on behalf of Riddington and Rimmer (Birkdale Garages) Ltd. Southport.*

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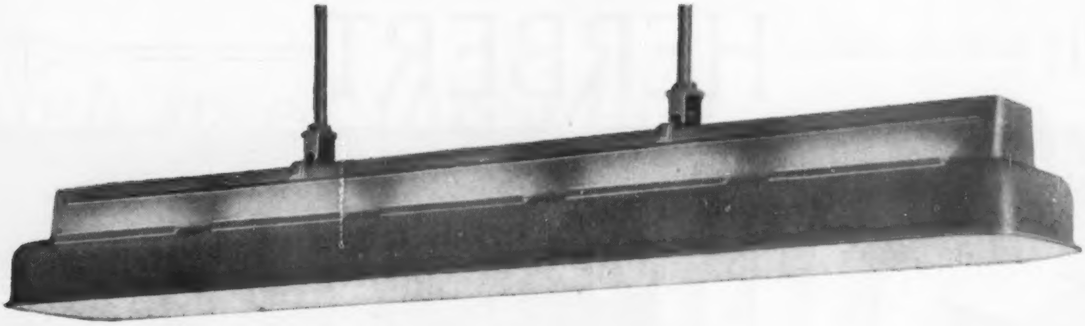

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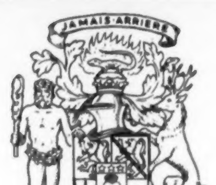

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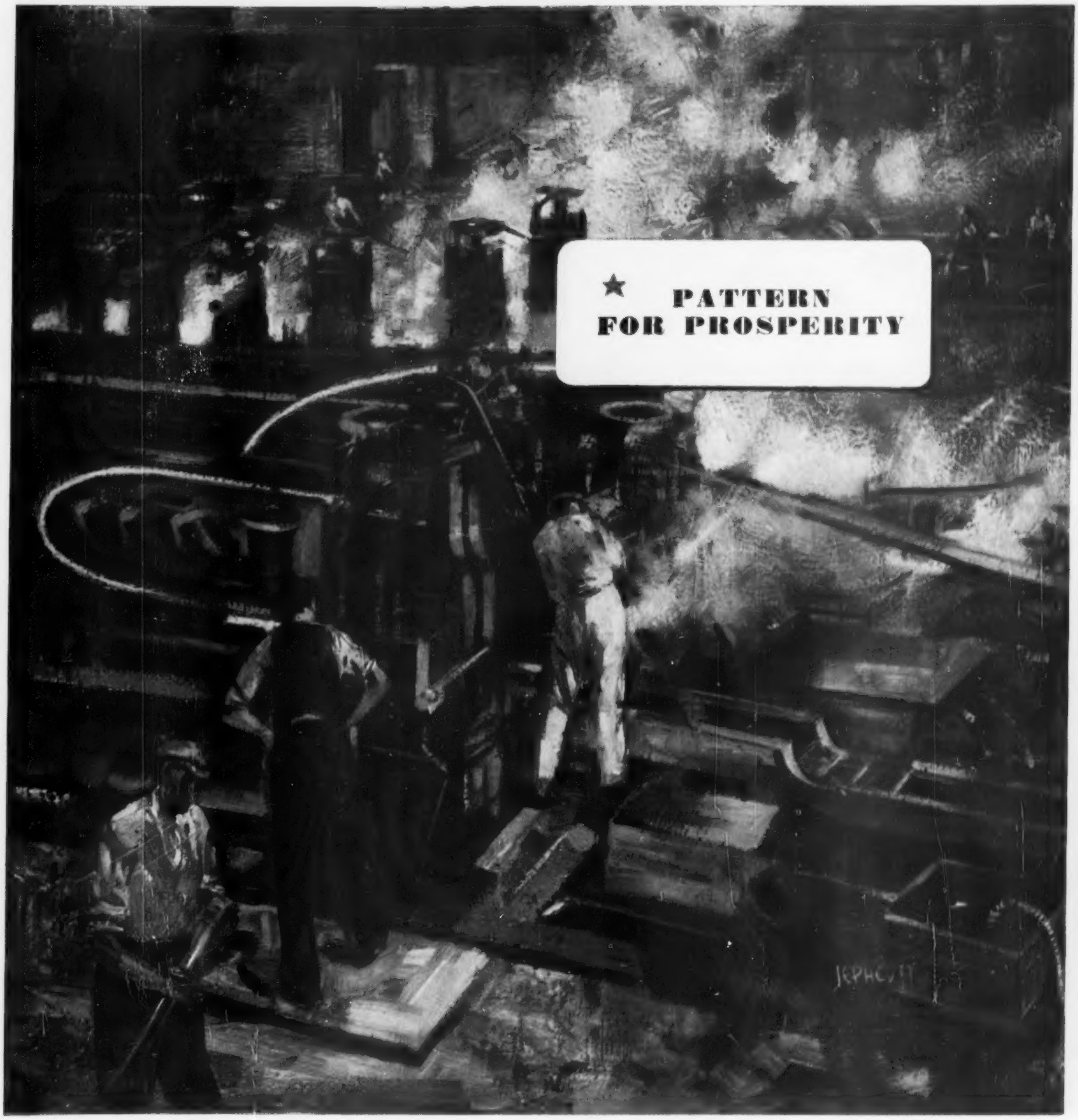
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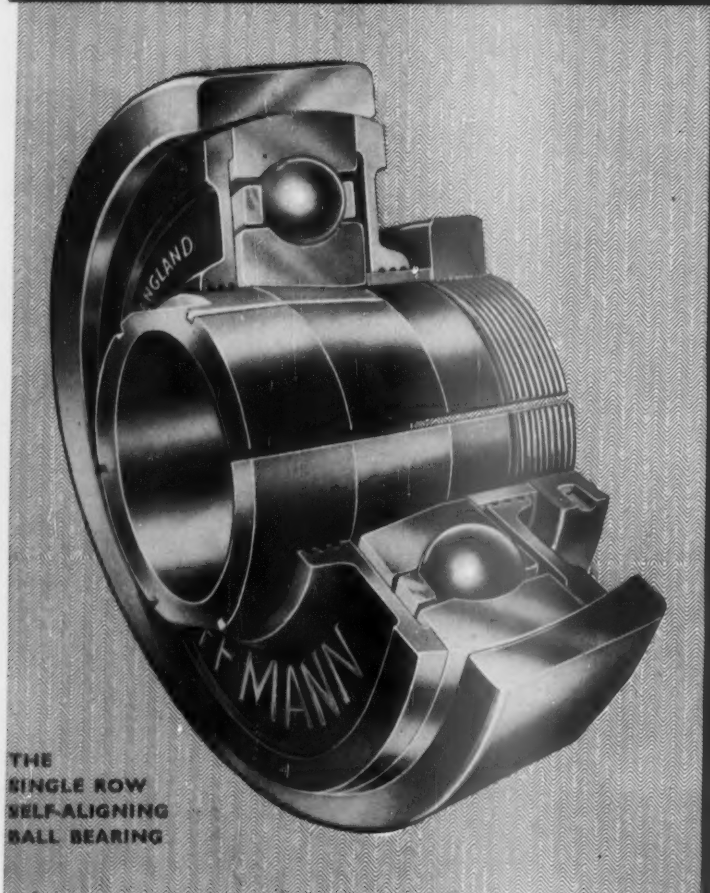

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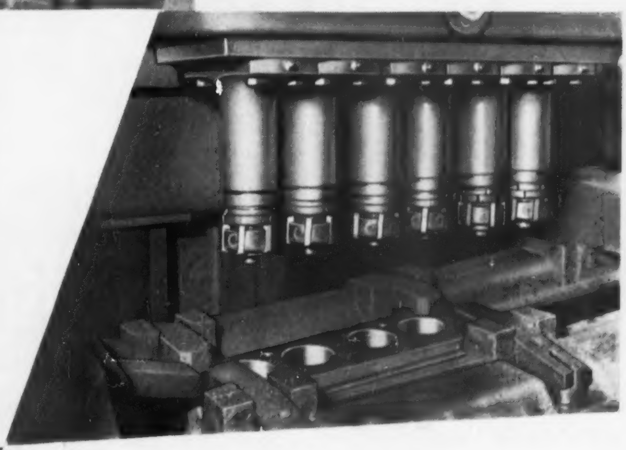

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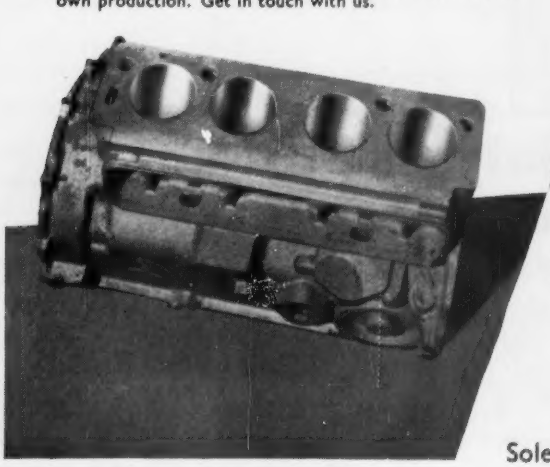

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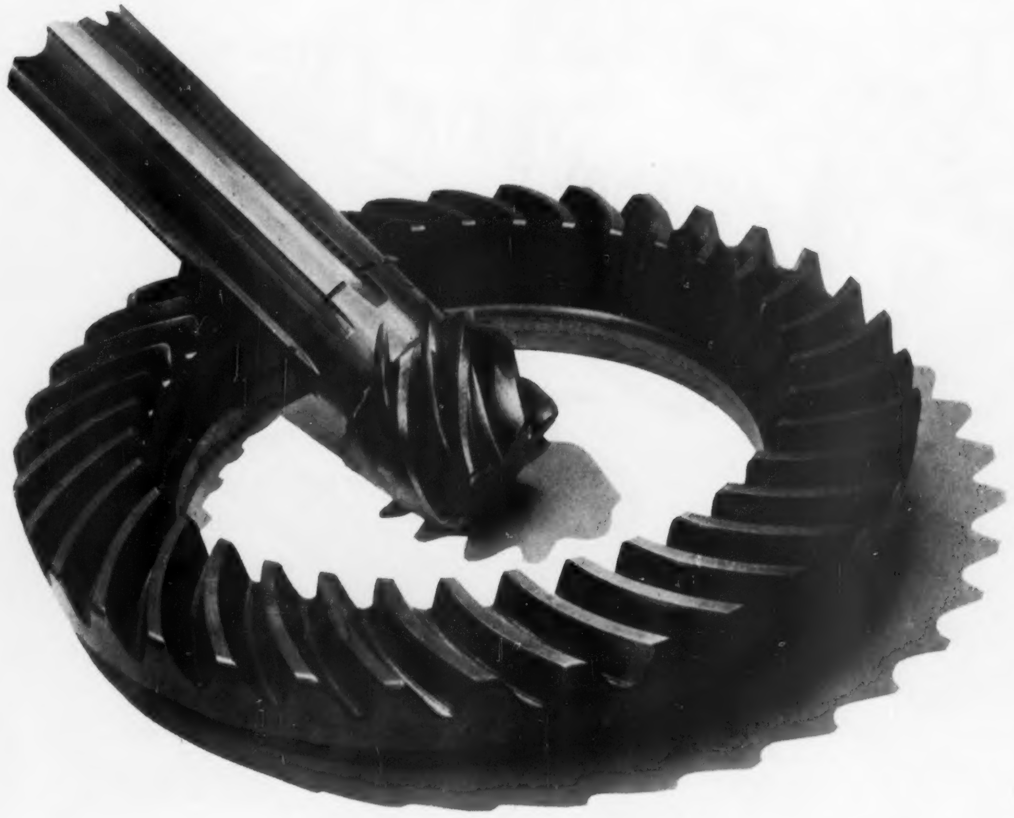

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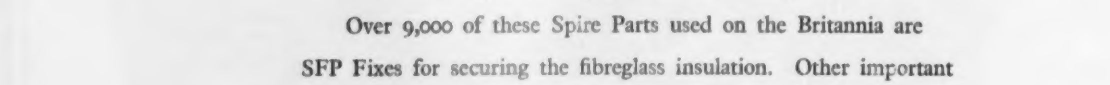

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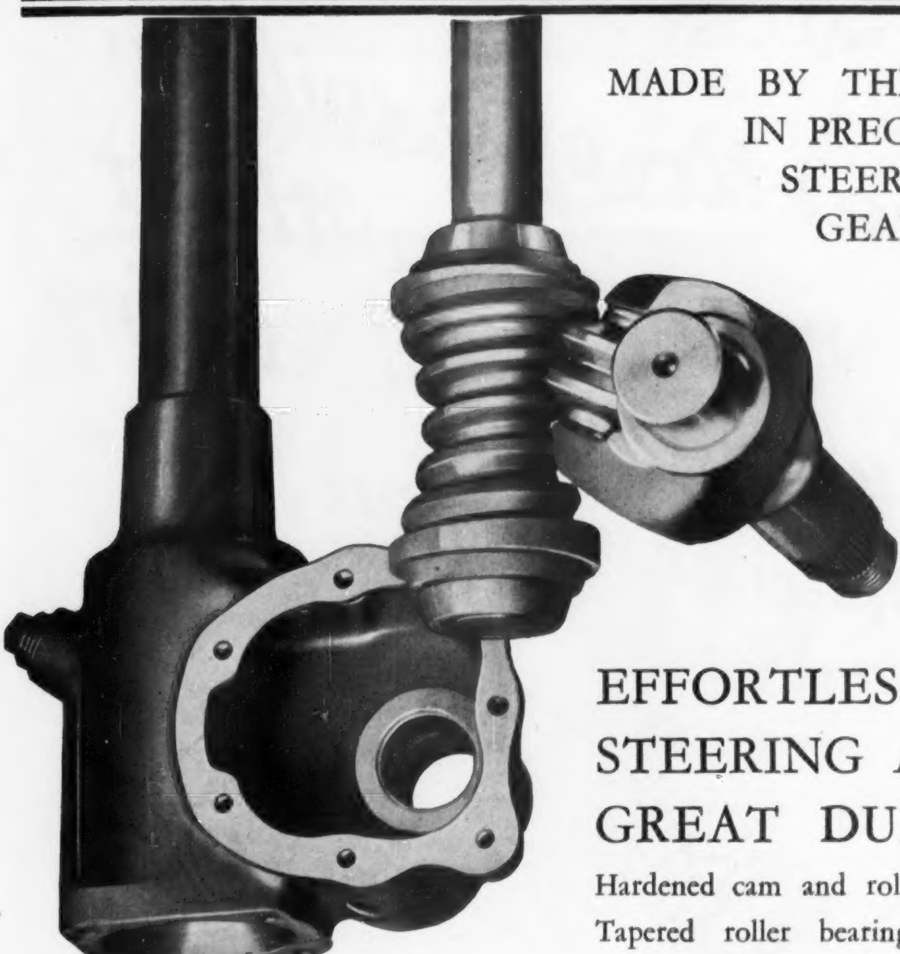

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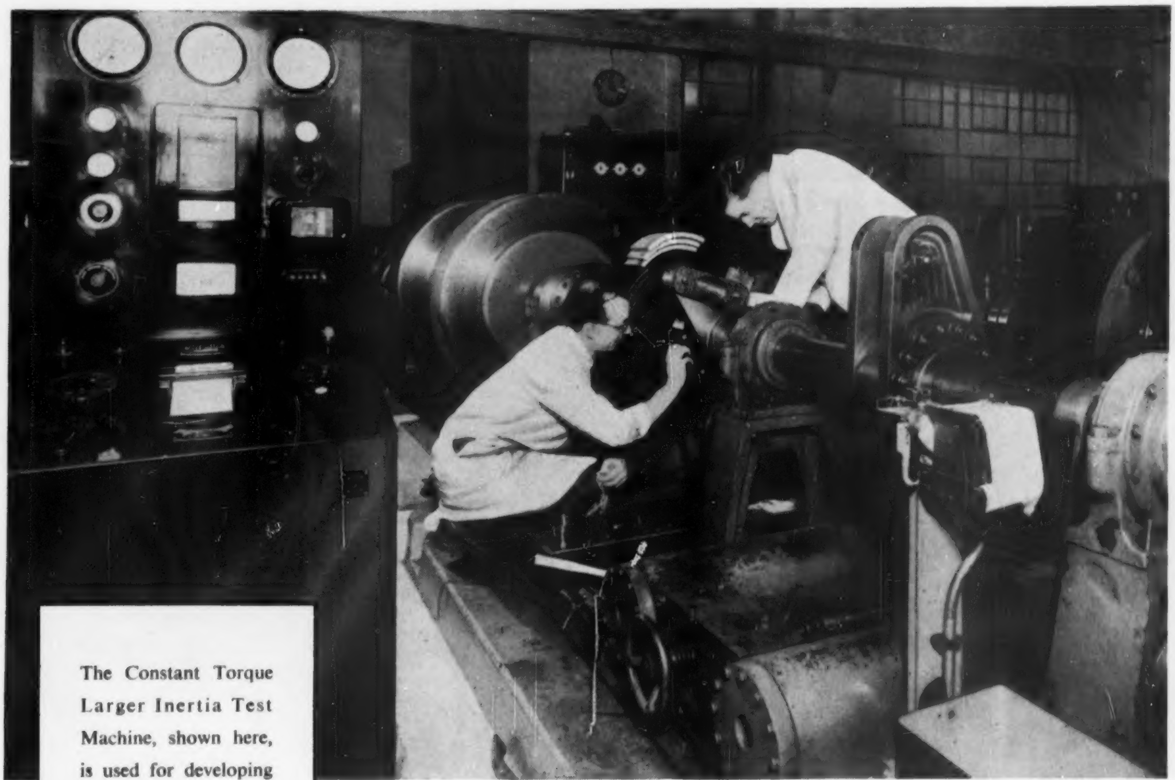

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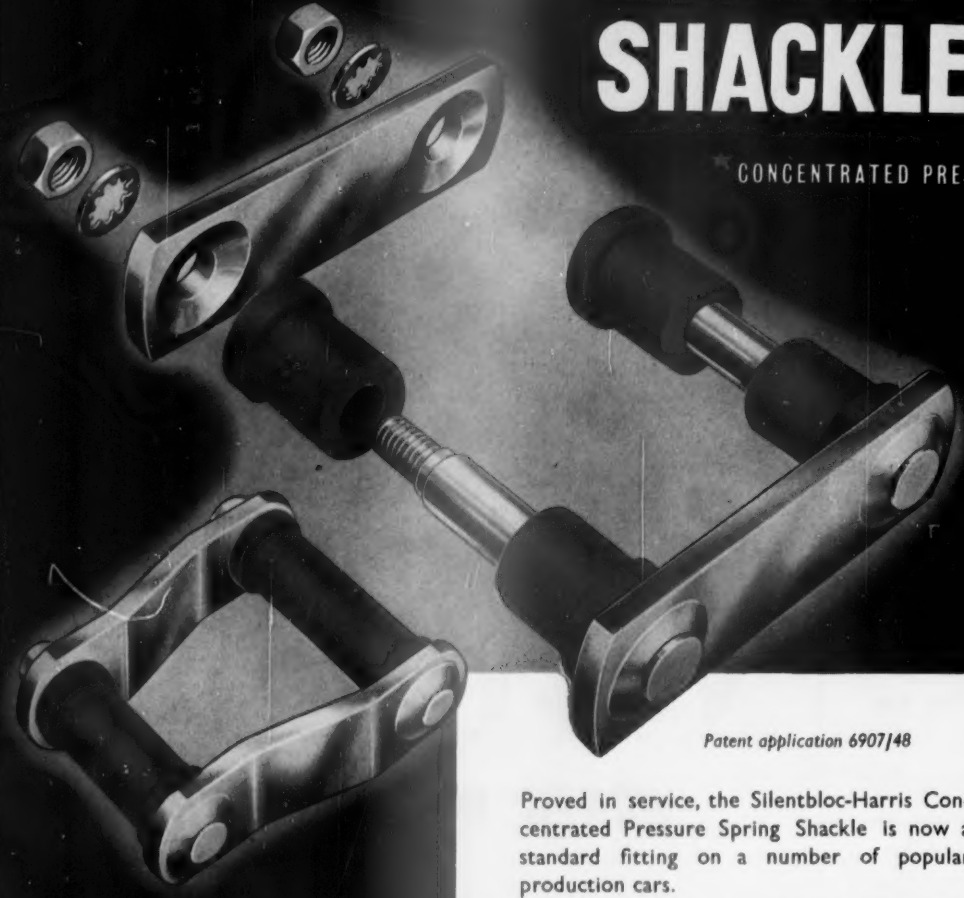

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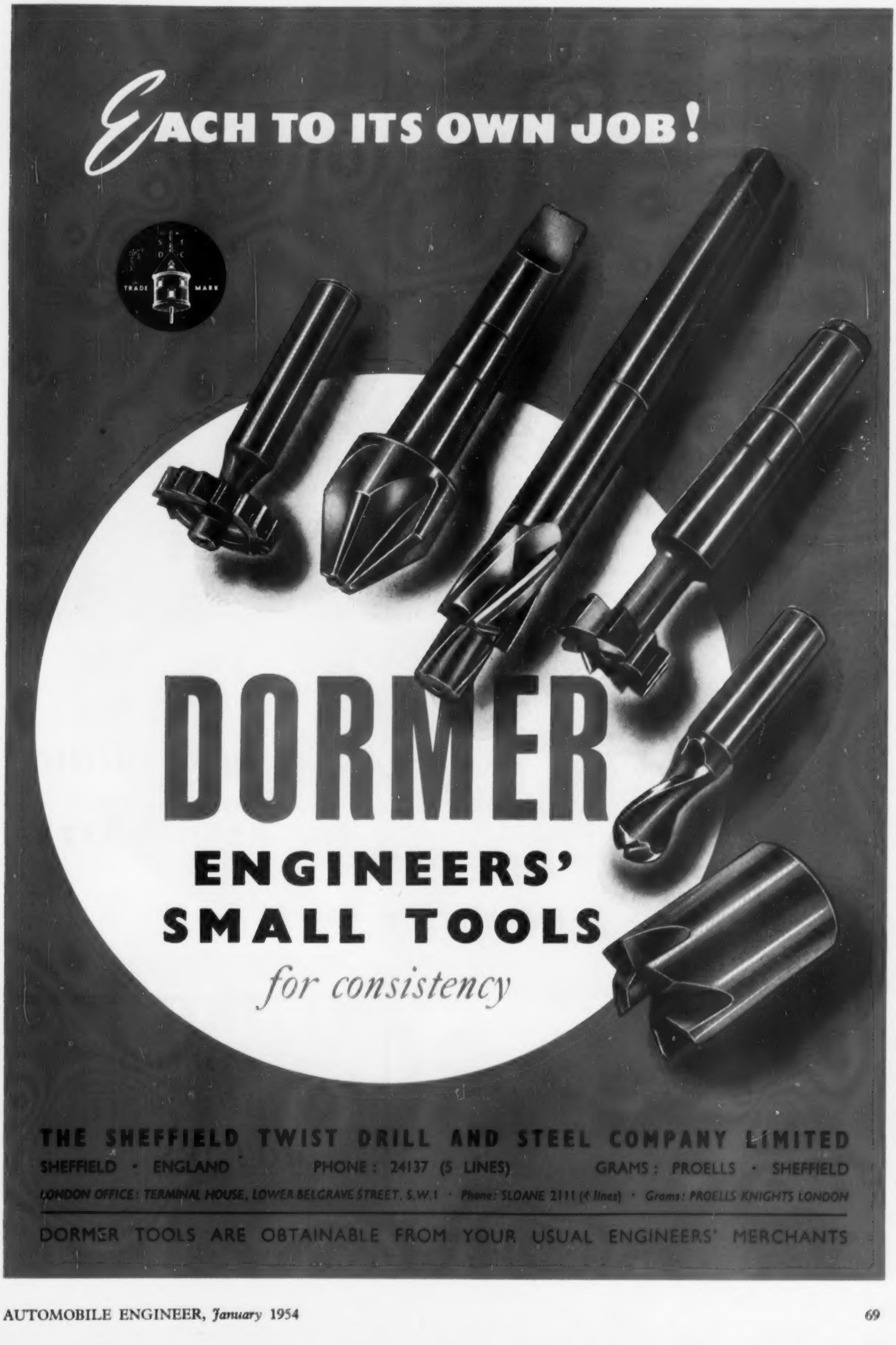
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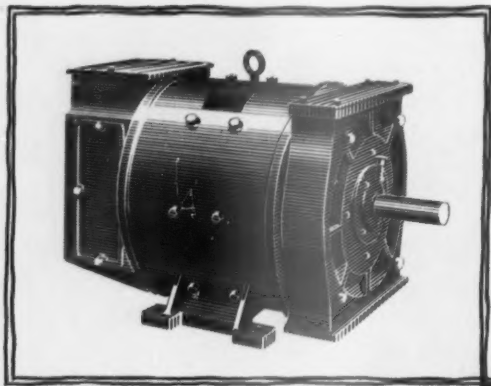

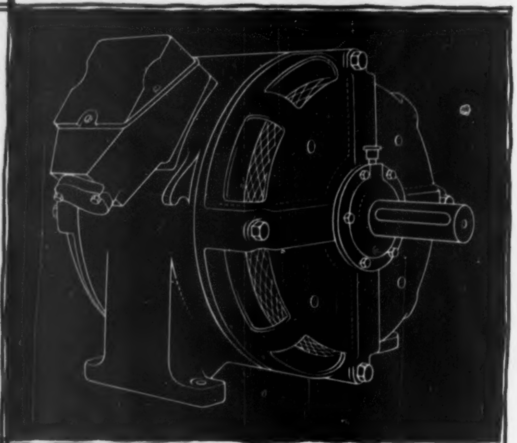

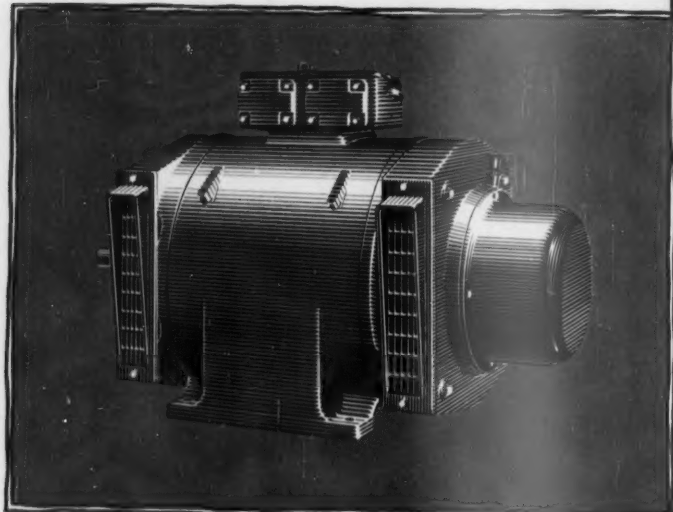

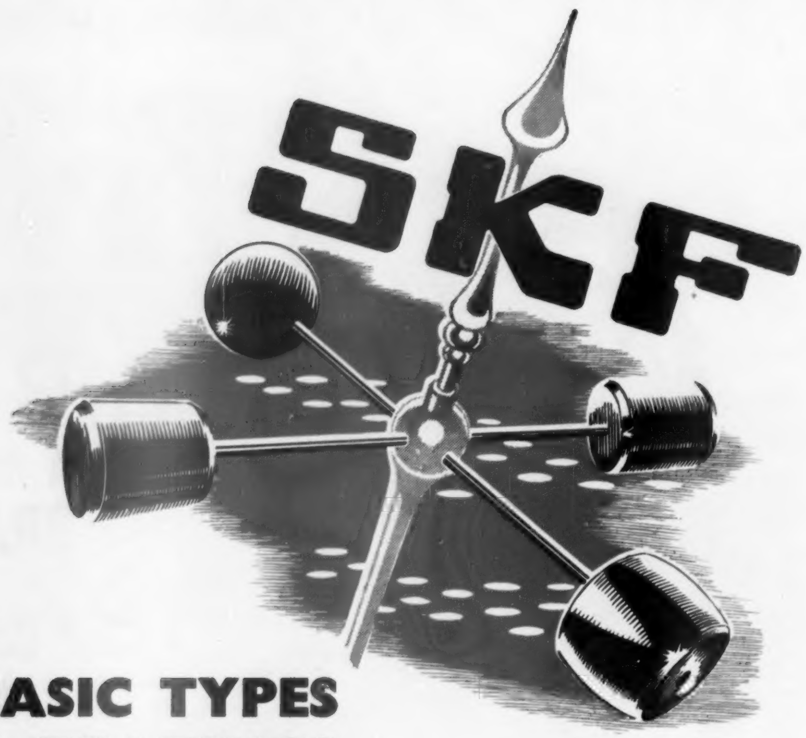

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Commercial Vehicle Engines

SINCE the war there has been a marked increase in the demand for diesel engines as power units for heavy and medium-size commercial vehicles. Of the few manufacturers who still make petrol engined trucks, almost all offer a diesel installation as an alternative. This trend has developed mainly because of the high cost of fuel in many parts of the world. It is even apparent in countries where the rates of taxation on petrol and diesel oil are such as to make both approximately the same price, for the low brake specific fuel consumption characteristic of compression ignition engines leads to appreciable economy in operation over large mileages.

Under the present conditions—which could change—the petrol engine only retains its precarious footing in the commercial vehicle market because of its low prime cost. The difference between the price of these units and that of diesel engines is by no means all due to the relatively low cost of ignition equipment as compared with injection system components. In fact, much of the price differential in favour of spark ignition types arises, in some cases, because the power units employed are common to private cars as well as to trucks, and are therefore manufactured in much larger numbers than are the diesel units designed for trucks only.

Small vehicles

During the past year or two, much attention has been devoted to the development of diesel units for taxis and private cars. Production in the quantities needed to meet the demand for these two applications alone is unlikely to be adequate to warrant designing an engine specifically for these light vehicles. Yet, if this type of power unit is to give complete satisfaction in private cars, it must be of a specialized design.

To meet this requirement, attention is likely to be turned towards light commercial vehicles, with a view to widening the scope for the development of diesel engines of this class. Many of these vehicles are built on chassis of private cars and a diesel engine might be offered as an alternative to the normal petrol engine in both. The difficulties to be overcome to make such an arrangement practicable are considerable but not insuperable. If both the petrol and the diesel units develop the same maximum power at not greatly differing speeds, and have similar torque characteristics, transmission design is simplified. That high speeds are practicable in diesel units has been demonstrated by a number of manufacturers. One of the Cummins diesel engines, developed for racing, produced its maximum b.h.p. of 340 at 4,000 r.p.m., and the output of the Borgward unit is 42 b.h.p. at 3,400 r.p.m. Experimental work by the Caterpillar Tractor Company is said to have demonstrated that a small bore, air-cooled diesel engine can be run at speeds up to 6,000 r.p.m. without undue sacrifice of economy.

Heavier vehicles

Air-cooled diesel engines have been well received in certain markets. They are particularly suitable for operation under conditions of extreme cold, and it is not surprising, therefore, that the Swiss Locomotive and Machine Works is one of the manufacturers who have introduced engines of this type. Another air-cooled series that has been running for some time is the Magirus Deutz range of engines.

Success in air-cooling is largely dependent on the design of an efficient blower and baffling to distribute the blown airstream round the cylinders. It has long been realized that intake duct shape and blade profiles and angles are critical and that much can be done to eliminate noise by incorporating guide vanes to direct the air at a suitable angle on to the impeller blades. When all this has been done there can be but little difference between the costs of air- and water-cooled engines. However, it is claimed that air-cooled units can be made lighter.

If this is so, they conform with the general trend towards all round weight reduction to increase the ratio of pay load to vehicle weight. Nevertheless, they are not likely to be well received in this country, because they are generally noisy and therefore not entirely acceptable for passenger transport vehicles. Moreover, we are not troubled with extremes of climate, and it is unlikely that manufacturers will ever design solely for export, even to dollar countries.

The world trend towards higher cruising speeds has, of course, necessitated the adoption of high-speed engines of greater power. However, because of the increased weight and size of these engines, their installation has tended to reduce the load carrying capacity of vehicles designed to comply with legal restrictions on all-up weights and dimensions; so to offset this tendency, more attention is being devoted to weight reduction and compactness of power units. Gas turbines, because of their high power: weight ratio and small size, are an attractive alternative

to diesel engines, but obviously their specific fuel consumption will have to be drastically reduced if they are to become acceptable in most parts of the world.

Supercharging, as an alternative means of obtaining a high power output from an engine of relatively small size, is often adopted in mountainous areas, where a high power:weight ratio is generally regarded as essential to maintain high average speeds. Roots-type blowers are usually employed, but they have their limitations, and their optimum discharge pressure is generally accepted as being about 7½ lb/in². The centrifugal type is not easily adapted to road vehicle engines because of the wide ranges of load and speed involved. It would appear that exhaust-driven turbo-superchargers might be further developed to make them suitable for application to road vehicles. With these, the engine torque falls off rapidly as the speed is reduced; but this unfavourable characteristic might be eradicated by incorporating nozzles of variable size in the turbine unit that drives the supercharger.

In this country, the continued restriction on commercial vehicle speeds prohibits manufacturers from following the world trends, and inevitably will increasingly effect our ability to compete in overseas markets. Although the use of more powerful engines would give higher speeds on uphill gradients and better acceleration in traffic, long distances on level ground would have to be covered at uneconomically small throttle openings. Moreover, so long as the maximum permissible speed is 20 m.p.h., or even 30 m.p.h., there will be little scope for reduction in journey times. Admittedly, an increase in the maximum speed allowed would involve expenditure on roads to make them capable of carrying the faster traffic, but the march of progress cannot for long be halted and in halting it even for a relatively short period, there is considerable risk to the economy of the country.

Stability

DURING the history of the world's motor industry, private car models have been produced from time to time that have had a tendency to overturn during high-speed cornering. Three design features largely determine whether or not a model will have this characteristic. One is the wheel track, the second is the roll stiffness, and the third is the height of the centre of gravity of the vehicle. Lack of roll stiffness leads during cornering to a movement of the centre of gravity towards the outer pair of wheels and, therefore, to instability. On the other hand, complete absence of any tendency to roll can also be dangerous in that little warning is given to the driver to indicate that he is cornering too fast.

Other features also have an effect on the overturning tendency; for instance, the wheelbase, aerodynamic characteristics, and the position of the centre of gravity relative to the longitudinal and lateral axes of the vehicle. Nevertheless, it seems more than likely that an empirical rule, taking into account only the three factors mentioned in the previous paragraph, could be formulated for the guidance of designers, so that they can avoid introducing features that may lead to overturning. Moreover, the data collected for such an investigation would also be useful as a guide to suspension design. The need for, and stiffness of, anti-roll devices must be determined largely by the height of the centre of gravity of the sprung mass above the roll centre.

The risk of overturning is, of course, greatest in small cars because, in order to economise in space and materials, both the track and the wheelbase are generally reduced as much as practicable. When the wheel base is short, it is often necessary for the rear seat pan to be relatively high so that it will clear the nose of the back axle and, in some cases, to ensure that the rear passengers' elbows shall be clear of or rest comfortably on the tops of the wheel-arches. This tendency to raise the centre of gravity is generally countered by increasing the width of the wheel track.

In view of the importance of the position of the centre of gravity of a vehicle as a factor affecting a number of its handling characteristics, there is every reason to be surprised that many manufacturers can only quote the position, relative to the wheel axes, of the transverse vertical plane in which the centre of gravity is. If records of the exact positions of centres of gravity as well as of the relevant handling characteristics of all models had been kept in the past, a mass of data on which to base future designs would now be available. However, selective testing of old-type cars known to have overturning tendencies and a wide variety of other cars with varying characteristics, could provide the necessary information. It might also be useful to know the magnitude of the lateral force, applied horizontally through the centre of gravity, necessary just to overturn the vehicle.

CONTENTS

ARMSTRONG SIDDELEY SAPPHIRE CHASSIS

A Radical Departure from Previous Armstrong Siddeley Practice

IN the past, Armstrong Siddeley cars have not been in the high performance class, but rather have they been noted for their solid construction and reliability. However, modern developments and design techniques have made it possible to reduce the weight of motor vehicles generally without adversely affecting strength, stiffness or reliability. Moreover, because of the high cost of fuel in this country, and in many others, a low consumption in terms of miles per gallon is required by most customers. Therefore, weight must be kept to a minimum.

To meet these requirements, the aim of the manufacturers of the Sapphire has been to produce a car with a high power:weight ratio and capable of high top speeds, but without making any sacrifices so far as dependability or comfort and ease of handling are concerned. The car cruises comfortably at 75-80 m.p.h. and, with a single carburettor fitted, its top speed is over 90 m.p.h., while that of the twin carburettor version is over 100 m.p.h. An outstanding feature of the vehicle, and one which reflects considerable credit on the designers, is its relatively low cost. This feature has been attained without any lowering of general standards and should help considerably to enable the vehicle to compete favourably in world markets.

The principal dimensions of the Sapphire, which is powered by the 3·4 litre engine described in the December 1953 issue of *Automobile Engineer*, are given in the specification panel. From this, it can be seen that the dry weight is 3,472 lb and, since with an engine with a single carburettor the b.h.p. developed is 125, the power: weight ratio is 80 b.h.p/ton; with the twin carburettor version, which develops 150 b.h.p., the ratio is 97 b.h.p/ton. When the vehicle is empty, the front/rear weight distribution is 50/50. The weight of the body in the white is 935 lb. It is of interest to compare the weights of the two gearboxes, each with its appropriate flywheel and topped up with oil. The synchromesh unit weighs 202 lb, while the preselector gearbox weighs only 188 lb.

SPECIFICATION

TRANSMISSION: With the preselector gearbox, an Armstrong Siddeley centrifugal, $9\frac{1}{2}$ in diameter, single dry plate clutch is used. Preselection is effected electrically. Gear ratios: top 1:1, third 1·36:1, second 1·993:1, first 3·4:1, and reverse 4·75:1. With the synchromesh gearbox, a Borg and Beck, 10 A6 G, 10 in diameter, single dry plate clutch is employed. Gear ratios: top 1:1, third 1·42:1, second 2·09:1, first 3·13:1 and reverse 3·31:1. A Hardy Spicer open-type propeller shaft with an intermediate bearing is used in conjunction with each gearbox. Rear axle: Salisbury, semi-floating type with a hypoid pinion, and a final drive ratio of 4·091:1.

FRONT SUSPENSION: Double transverse wishbone-link system, with coil spring and anti-roll bar, and Girling, DAS 6 telescopic shock absorbers.

REAR SUSPENSION: Semi-elliptic leaf springs with a through axle and anti-roll bar, and Girling DAS 8 telescopic shock absorbers.

STEERING: Burman recirculatory ball type, with a ratio of 22:1 giving 3 turns from lock to lock. Turning circle 42 ft 6 in.

BRAKES: Front, Girling hydraulic HLS/S, two leading shoe type. Rear, Girling hydraulic HNS/S/H, leading and trailing shoe type. Drum diameter 11 in. Shoe width $2\frac{1}{4}$ in. Total friction area 184 in^2.

TYRES: 6·70 × 16·00 on $5\frac{1}{2}$ in wide rims. Pressures, 24 lb/in^2.

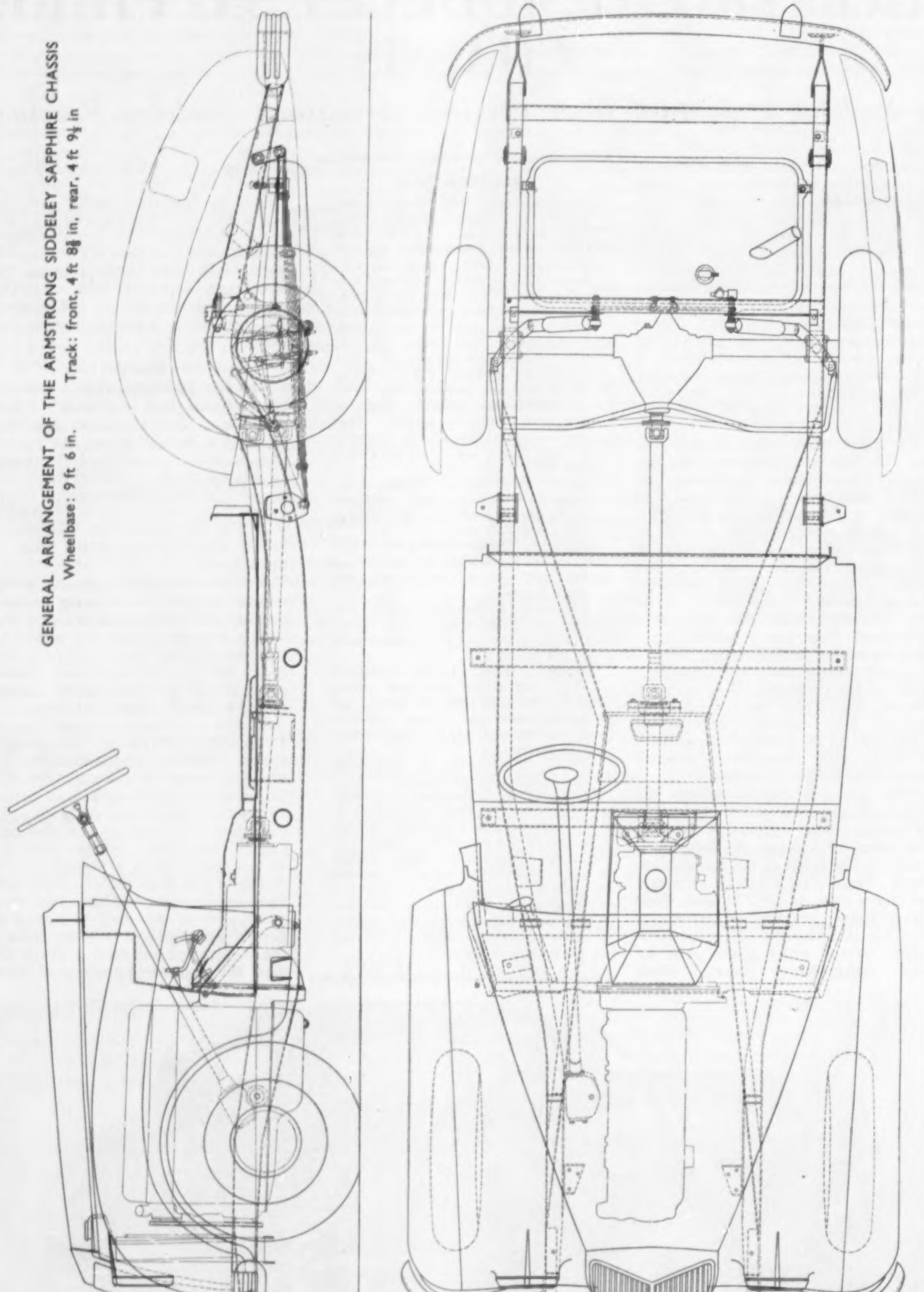
DIMENSIONS: Wheelbase 9 ft 6 in. Track, front, 4 ft $8\frac{5}{8}$ in, rear, 4 ft $9\frac{1}{2}$ in. Ground clearance 8 in laden. Overall length 16 ft 1 in. Overall width 6 ft. Overall height 6 ft 3 in laden. Frontal area 25 ft^2. Dry weights: vehicle 3,472 lb; front/rear weight distribution 50/50; engine, less flywheel, clutch and gearbox 545 lb; gearbox, with flywheel and oil, preselector 188 lb, synchromesh 202 lb; body (white) 935 lb.

Clutch

The clutch employed when the synchromesh gearbox is fitted is a Borg and Beck, 10 A6 G, single dry plate unit. It is 10 in diameter, the friction area is 81·6 in^2, and twelve pressure springs are fitted. A carbon thrust ring is used in the disengagement mechanism. The whole unit is enclosed in a D.T.D.424 aluminium bellhousing bolted and spigoted to the front of the gearbox.

With the preselector gearbox, a centrifugal clutch of Armstrong Siddeley design and manufacture is used. It is a $9\frac{1}{2}$ in diameter single dry plate unit, and the friction area is 85 in^2. This clutch comprises a centre plate, presser plate, springs and thrust plate, together with the bob weights and levers, and the whole assembly is enclosed in a cast D.T.D.424 clutch casing. The casing is spigoted into the rear face of the flywheel, to which it is secured by nine, $\frac{3}{8}$ in diameter set bolts. The six presser springs, 8 S.W.G. by 0·875 in inside diameter, each bear at one end in a circular recess in a presser plate and at the other end in a flanged thimble fitting in a hole in the $\frac{3}{8}$ in thick, mild steel thrust plate. Three shouldered studs screwed into the rear face of the presser plate project through holes in the thrust plate and carry on their rear ends the stop nuts by means of which

The Armstrong Siddeley Sapphire has an exceptionally pleasing appearance, and all round visibility is good



GENERAL ARRANGEMENT OF THE ARMSTRONG SIDDELEY SAPPHIRE CHASSIS
Wheelbase: 9 ft 6 in. Track: front, 4 ft $8\frac{3}{8}$ in, rear, 4 ft $9\frac{1}{4}$ in

the initial compression of the springs is adjusted. These nuts are locked by split pins. The free length of the springs is 2·75±0·005 in, their fitted load is 163 lb, and the rate is 260 lb/in.

Rotation of the thrust and presser plate assembly relative to the casing is prevented by three tie plates, each of which is secured by two $\frac{5}{16}$ in diameter nuts on studs in the rear face of the presser plate. The tie plates are held clear of the presser plate by distance pieces, and trail relative to the direction of rotation of the engine. Their other ends are carried on pedestals, which are mounted on the clutch casing and which project through $\frac{11}{16}$ in diameter holes in the thrust plate. Each pedestal consists of a $\frac{3}{8}$ in diameter centre rod which is free to slide axially in a circular section outer portion. A nut, together with a distance piece, pulls each tie plate against a collar machined round the front end of the centre portion, the rear end of which is threaded for two nuts locked together to limit the sliding motion. Round the $\frac{9}{16}$ in diameter periphery of the outer component of the pedestal and approximately mid-way between its ends is a collar. The outer component is spigoted into the front face of the clutch casing and secured by a nut tightened against the rear face.

When the engine is stationary, or only turning slowly, the clutch is disengaged, that is, the presser and thrust plate assembly is pushed to the rear by three compression springs, 0·390 in inside diameter by 8 S.W.G. The front ends of these springs are carried in flanged thimbles in holes in the flywheel. Each spring projects through a large hole in the presser plate and its rear end bears against a shoulder on an adjuster bolt. The end of this bolt is not threaded and projects into the spring which it thus locates radially. The bolt is screwed through a tapped hole in the thrust plate, and its setting, which determines the rate of take-up of the clutch, is fixed by means of a lock-nut tightened against the plate. A fitted load of 103 lb is specified for the spring, which has a free length of 2·315±0·05 in and a rate of 740 lb/in. Rearward travel of the thrust and presser plate assembly is limited by three stops spaced 120 deg apart. These stops consist of an inner and an outer portion. The outer part is in the form of a hollow bolt screwed into the clutch casing from the rear and locked by a tab washer. Screwed into it axially is the bolt which serves as the stop, the setting of which is fixed by a lock nut. This arrangement is necessary to avoid any possibility of damage to the threads in the aluminium clutch casing during adjustment.

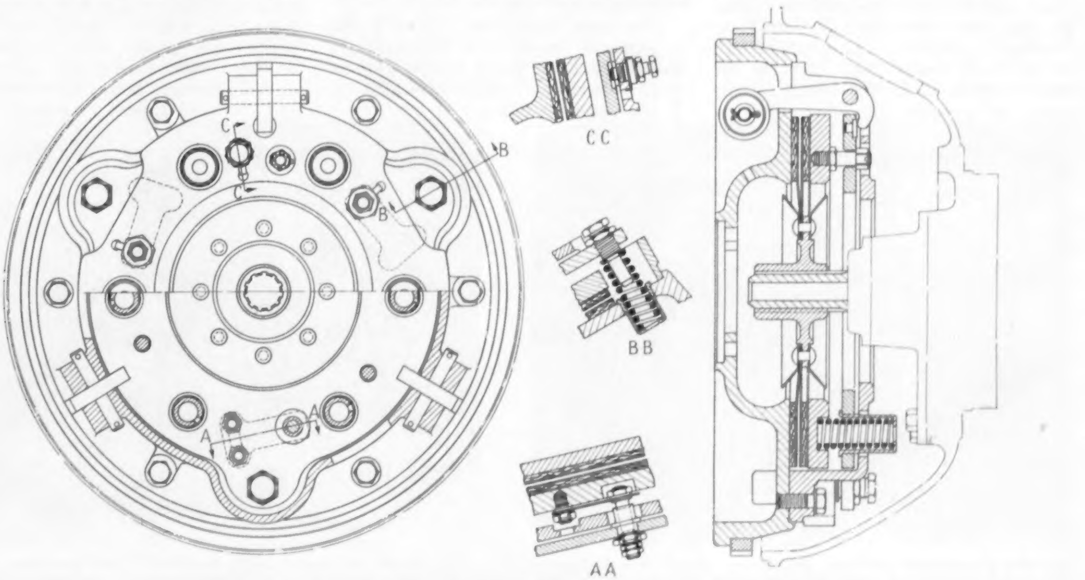
Three bob weights are carried inside the rim forward of the flywheel disc. Each is secured by a $\frac{3}{8}$ in diameter bolt to the front end of a case hardened En 32 lever which projects through the flywheel disc and is carried, between two lugs on the clutch casing, on a $\frac{7}{16}$ in diameter, En 6 pivot pin. This pin is retained by a split pin at each end. The rear ends of the three levers are turned through 90 deg immediately to the rear of their pivot pins and bear on circular pads, of a chrome carbon steel similar to En 31, which are shouldered and pressed into holes in the thrust plate.

As the engine speed is increased, the weights are flung outwards by centrifugal force and the levers move the thrust plate forwards. This, acting against the disengagement springs, pushes the thrust and presser plate assembly forward until it comes up against the centre plate of the clutch. Initial engagement is effected gently by virtue of a special axially sprung centre plate arrangement. Forward motion of the thrust plate relative to the presser plate, and therefore the compression in the springs, is limited by the hexagon head formed near the base of each of the three studs that are used to set the pre-load in the presser springs. Initial take-up of the clutch occurs at about 675 r.p.m., and full torque can be transmitted at about 1,000-1,200 r.p.m. The weights have rubber rings fitted round their peripheries, and when they contact the inside of the flywheel rim, which is at about 1,400 r.p.m., these rings act as buffers.

Two friction facings are employed and they are carried one on the front and the other on the rear of the centre plate. This plate comprises two 20 S.W.G. dished, mild steel pressings placed back to back, together with two dished thrower rings, and secured by eight $\frac{1}{4}$ in diameter rivets to the flange round an En 100 splined boss. The thrower rings prevent any possibility of grease being flung outwards on to the thrust faces. When the clutch is engaged, the two dished plates are pressed flat, but they are not strained beyond the elastic limit so they spring back again when the clutch is disengaged.

Gearbox

Either a conventional, synchromesh gearbox or the Wilson-type unit with an electrically preselected mechanism may be fitted. The dry weight of the synchromesh unit, complete with bell-housing, clutch withdrawal mechanism, connecting link to pedal, rear mounting bracket and companion flange for the propeller shaft universal joint, is 133 lb. With this gearbox, which is the same as that employed in the Humber Super Snipe, and described in the July 1953 issue of *Automobile Engineer*, the ratios are: top 1:1, third 1·42:1, second 2·09:1, first 3·13:1 and reverse 3·31:1. The gear shift lever is on the steering column.



A centrifugal clutch is used in conjunction with the preselector gearbox

Four forward speeds and one reverse are again obtainable with the pre-selector unit, which is the one that will be described. The ratios are: top 1:1, third 1·36:1, second 1·993:1, first 3·4:1 and reverse 4·75:1. Instead of the camshaft arrangement, which is generally employed in Wilson-type units to actuate the struts, a system of bell cranks, push rods and solenoids is used. This has made possible the adoption of a light gate type control on the steering column. The control is a switch by means of which the driver selects the circuit of the appropriate solenoid. However, the circuit is not completed until the gear shift pedal is depressed and a lever on the end of the cross shaft that actuates the spring gear contacts a micro-switch. Thus the load on the battery is very small. The reason for the adoption of this electrical control is that, unlike mechanical controls, it does not develop lost motion or other defects resulting from wear during service.

The gearbox casing is of D.T.D.424 aluminium, as also are all of its main covers. A D.T.D.424 aluminium bell-housing, formed integrally with the front cover and pump casing, is spigoted into the front end of the gear-box and secured by nine $\frac{5}{16}$ in diameter studs and nuts locked by spring washers. A transverse web, about $8\frac{3}{4}$ in from the front spigot face of the box, separates the running gear and spring gear compartments. The overall length of the unit from the front end of the primary shaft to the rear of the companion flange for the universal joint at the back is 22 in, while from the spigot face to the same point at the rear it is $16\frac{1}{8}$ in long. Over the running gear compartment, the depth of the unit is $10\frac{1}{4}$ in, while its overall width is $14\frac{1}{2}$ in.

The rear end of the mainshaft is spigoted into the Vibrac V30 output shaft, and the whole assembly is carried on four ball bearings. One of these bearings is housed in the rear end wall of the box; another, also carrying the output shaft, is in the central transverse web, while the other two are positioned one in front and the other behind the oil pump at the front end. This two-bearing, front end support layout has been adopted in preference to spigoting the mainshaft into the tail end of the crankshaft, because it is found to lead to smoother operation of the clutch. When a spigot bearing in the crankshaft is used with a centrifugal clutch, there is a tendency for the splines of the centre plate assembly to bind slightly on the shaft. This is because of the build-up of tolerances, and deflection of the gearbox relative to the engine. When clutch engagement and disengagement is effected by the more positive pedal control used for conventional units, the alignment of the two shafts is not so critical.

The pump body is formed by the bellhousing and is closed at its front end by a D.T.D. 424 aluminium, extension casting spigoted and bolted on. Assembled into this casting from the forward end is the front ball bearing, the outer race of which is held against a shoulder by the spigot of a D.T.D. 424 front cover, which houses the Gits oil seal. The inner periphery of this seal bears on the $1\frac{1}{8}$ in diameter mainshaft. Immediately in front of the seal, the shaft is splined to carry the clutch centre plate assembly. These splines are copper plated to prevent fretting, and their root diameter is 0·95 in.

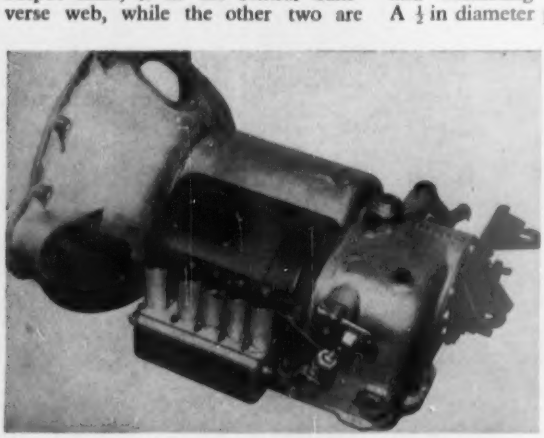
Behind the front bearing, the shaft is threaded to take a ring nut which is locked by a tab washer. This nut is tightened against an assembly comprising the eccentric of the oil pump, the inner race of the second ball bearing and the centre, or driving, member of the top speed clutch, all of which are pulled against a collar machined round the shaft. The ball bearing is passed from the front into its housing in the pump body and held in position by the spigot of the front extension which bears against the outer race.

The case hardened, En 32 pump eccentric is driven by a Woodruff key and actuates the usual hollow plunger and oscillating barrel arrangement. A $\frac{1}{2}$ in diameter plunger of cast iron is employed, and it is drilled axially [illegible] in diameter. It is integral with the ring in which the eccentric rotates. The barrel, which is circular in shape as viewed from the front, is of phosphor bronze and is 2 in diameter. The pump stroke is $\frac{1}{4}$ in.

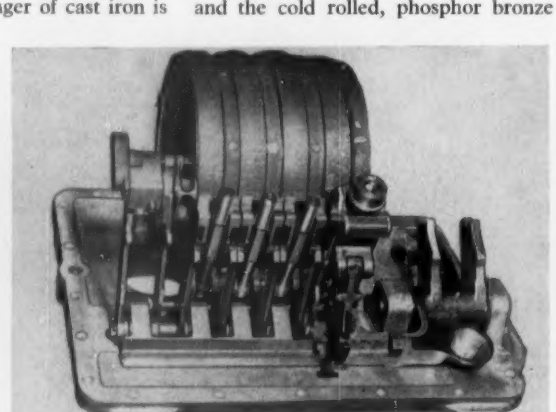
A noteworthy feature of the lubrication system is that the oil is drawn through a fine mesh strainer. This strainer is assembled, from below, into a chamber cored in the bottom cover of the gearbox. It is held against the $2\frac{3}{8}$ in diameter entry port, cored in the top of its chamber, by a coil spring compressed between the base of the filter and a cover plate bolted up to the bottom cover of the chamber. Oil passes through the filter and then out of the chamber past a ball type non-return valve. This valve is housed in the $\frac{3}{8}$ in diameter hole drilled vertically in the front wall of the gearbox, and its ball seats on an En 30 insert in the upper face of the bottom cover of the gearbox. A spring is not employed in the valve, but upwards movement of the ball is limited by a stop screwed in the top of the housing.

From the valve, the lubricant passes through a drilling in the front wall of the gearbox and another in the bell-housing, into the inlet port in the rear face of the pump barrel. Because of the oscillatory motion of the barrel, the inlet port is only in communication with the drilling in the body during the suction stroke. As the hollow plunger descends on the delivery stroke, the lubricant passes up to an annular space round the eccentric and thence, through a radial hole in the eccentric and shaft, into an axial drilling which passes the oil to the rear and distributes it through radial holes to the bush bearings in the sun wheels and tail end spigot.

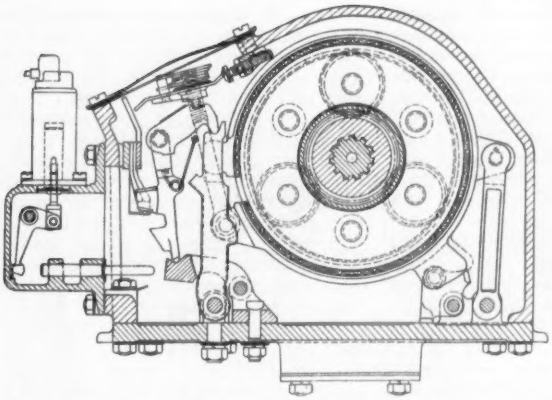
Running gear. The top speed clutch assembly is mounted on the case hardened En 32 centre member, already mentioned, which is driven by a plain key in the shaft. Four bright steel driving plates of the clutch are splined on the periphery of this centre member, and the cold rolled, phosphor bronze

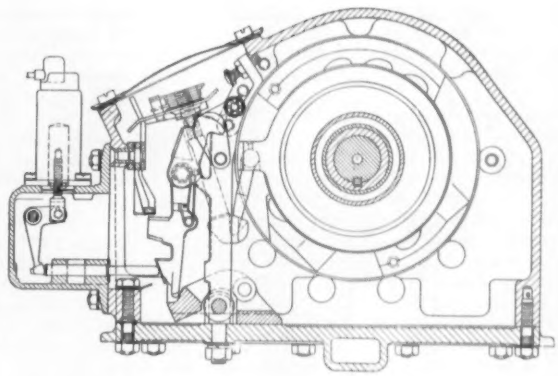


On the preselector gearbox, the micro switch that completes the circuit when the pedal is depressed is mounted behind the solenoids



The busbar, strut and toggle mechanism, and brake band assembly are shown here partly assembled on the bottom cover of the preselector gearbox





Left: A brake such as that used for first, second and third speeds, in the engaged position. Right: Fourth speed clutch actuating gear in the selected position

driven plates are splined into a forward extension of the third speed brake drum. The whole assembly is held together by two circular, mild steel plates riveted one in front of, and one behind the splined portion of the centre member. The rearmost driving plate bears against a phosphor bronze plate riveted to the front face of the clutch housing in the top speed brake drum.

To reduce the weight and therefore the inertia of the presser plate, it has been made of Hiduminium. It is carried on a phosphor bronze bush round the forward extended boss of the clutch centre member. A flange round this bush bears against a shoulder in the bore of the boss of the presser plate, against which it is held by a compression spring. The other end of this spring bears against the clutch centre member, and its function is to disengage the presser plate. In front of the presser plate, and carried on its boss, is a ball thrust bearing, the outer race of which bears on an En 8 conical thrust ring which seats in the inner of the two clutch actuating rings.

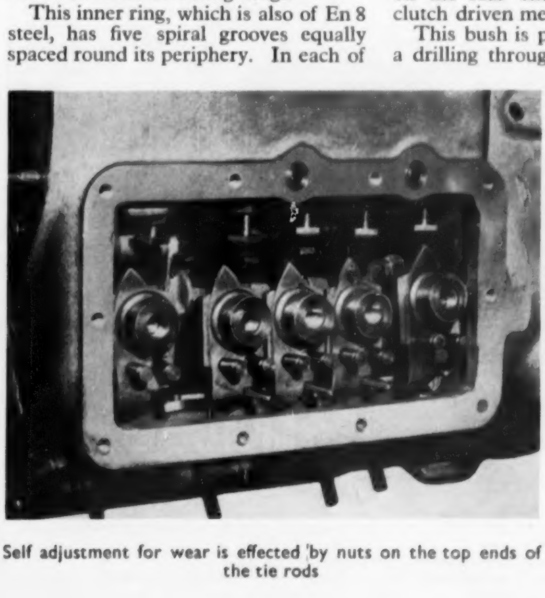
This inner ring, which is also of En 8 steel, has five spiral grooves equally spaced round its periphery. In each of these grooves is a ball running in a corresponding groove in the inner periphery of the outer ring, which is of high tensile, nickel alloy, cast iron. This outer ring is stationary and is spigoted in to the rear face of the bell-housing. It is located against rotation by two dowels, and against axial movement by five set screws countersunk radially into the periphery of the spigot. When top gear is selected, the centre member is rotated relative to the outer one, and the balls rolling in their spiral grooves cause it to move axially and engage the clutch.

The clutch driven member, which also serves as the third speed brake drum, is pressed on to the third speed sun gear, the front ends of the gear teeth being machined down to the pitch circle diameter to receive it. This sun wheel is carried on two flanged bushes, which are a light press fit in each end, and carried on the 1¼ in diameter mainshaft. The front portion of the planet carrier is formed by the second speed brake drum and annulus. It rotates on a short flanged bush, floating on the rear end of the boss of the clutch driven member.

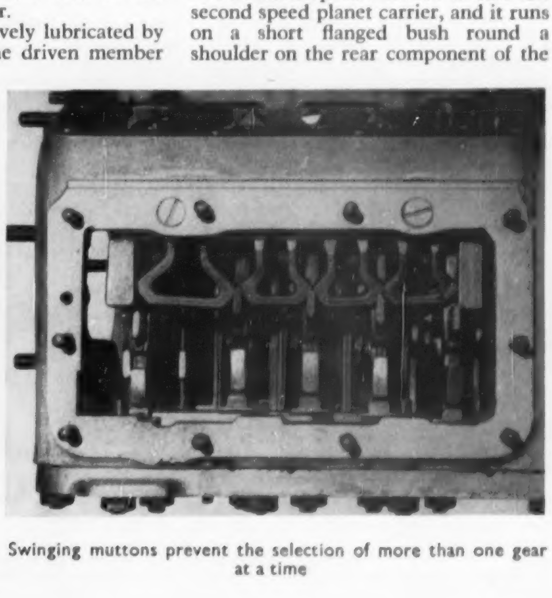
This bush is positively lubricated by a drilling through the driven member and sun gear to an annular groove round the inner periphery of the sun gear. Oil is fed into this annular groove, from another annular groove round the shaft, through holes drilled radially through the front sun wheel bush. This groove is fed with lubricant by a radial hole from the axial drilling in the shaft. The rear bush in the sun gear is served in a similar manner but, of course, there is no hole through the sun wheel at this point.

The rear component of the third speed planet carrier rotates freely on another short bush on the mainshaft. This bush again is lubricated through a radial hole in the shaft. Milled slots in the inner faces of both carriers transfer oil into annular grooves in the end faces of the planet gears. From these grooves, holes are drilled inwards to a slot milled longitudinally in the periphery of the journal. This lubricates the bearing surface of the 0·060 in thick Clevite bush, which is spherically indented to retain the oil. All the other planet gear bushes of the system are lubricated in a similar manner.

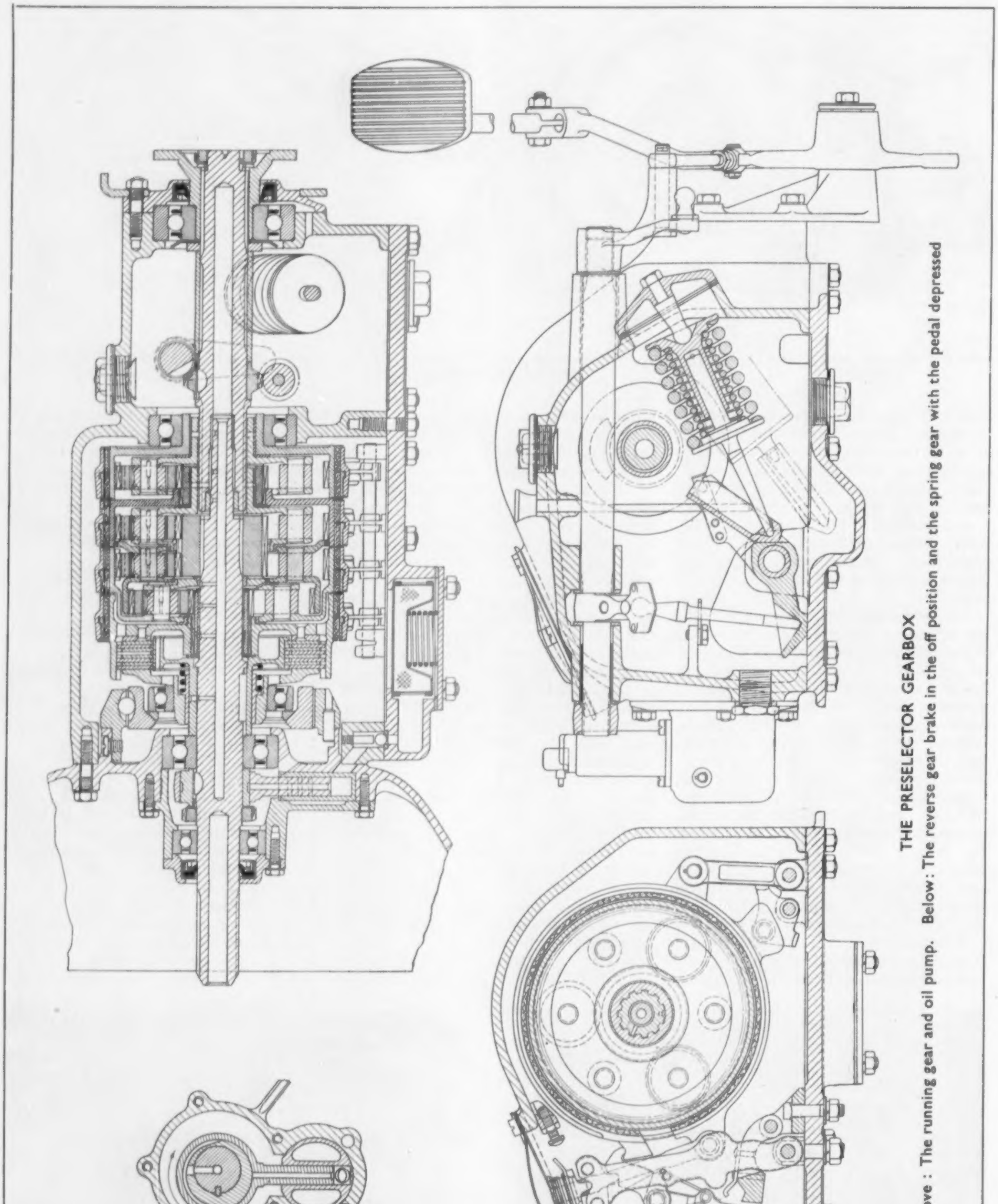
The third speed annulus is also the second speed planet carrier, and it runs on a short flanged bush round a shoulder on the rear component of the



Self adjustment for wear is effected by nuts on the top ends of the tie rods



Swinging muttons prevent the selection of more than one gear at a time



THE PRESELECTOR GEARBOX

Above : The running gear and oil pump. Below : The reverse gear brake in the off position and the spring gear with the pedal depressed

third speed planet carrier. A common sun gear serves both the second and the first speeds. It is driven by splines on the mainshaft which are cut to a root diameter of 0·95 in. The rear component of the second speed planet carrier is splined into the first speed annulus and brake drum, and it is carried on a flanged bush on a shoulder round the first speed planet carrier.

This carrier rotates clear of the sun wheel and is supported by the rear component, which is splined to the output shaft. Between the front face of this rear component and the sun gear is a phosphor bronze thrust washer, and round its rearward extended boss, on a fully floating flanged bush, is the reverse speed sun wheel. Pressed on to the front of this sun wheel, where it is machined down to the pitch circle diameter, is the first speed annulus and brake drum. The reverse annulus and brake drum is carried on a fully floating flanged bush round the boss of the rear component of the reverse speed planet carrier which also is splined to the output shaft, and which is separated from its sun gear by a thrust washer. Carried on the rearward extended boss of the reverse annulus and brake drum is the inner race of the ball bearing housed in the transverse web of the gearbox. The front component of the planet carrier rotates clear of the boss of the first speed annulus and brake drum.

With regard to material specifications, all planet and sun gears are of En 36V, cyanide hardened, the annuli and planet carriers are of En 110T, and the planet gear spindles are of case hardened En 32. The bushes and thrust washers are of phosphor bronze except, as has already been mentioned, the Clevite bushes for the planet gears.

Operation. From the foregoing description, it can be seen that the reverse speed sun wheel, first speed annulus and brake drum, second speed planet carrier, and third speed annulus are all locked together. This can be seen clearly in the illustration, in which these components are stipple-shaded. It follows that when the first speed brake is applied, the drive is taken through the first speed sun wheel splined to the mainshaft and thence by way of the planets to their carrier splined to the output shaft.

Should the second speed brake be applied, the drive is taken once again through the same sun wheel which, it will be remembered, engages both first and second speed planet pinions. This sun wheel rotates the second speed planets against their fixed annulus. Then, not only is the common sun wheel rotating at engine speed but the first speed annulus is also rotating at a different speed in the same direction, thus increasing the speed of the first speed planet carrier relative to that obtained when the first speed drum was held. So the drive is once again transmitted to the output shaft through the first speed planet carrier.

When the third speed brake is applied it locks the sun gear. The drive once again is taken from the mainshaft to the output shaft through the combined first and second speed sun wheel, the first speed planets and their carriers. Again the speed of the carrier is modified by that of the annulus, which is part of the system shown stipple-shaded in the illustration, and which under these conditions is rotated by the third speed sun gear and planets.

For top gear operation the clutch, engaged in the manner already described, locks the mainshaft to the third speed sun wheel. Then all the sun wheels, except the reverse one, are locked to the mainshaft. As is the case when each of the other speeds is engaged, the final drive to the output shaft must be taken through the first speed sun wheel, planets and planet carrier. The other sun wheels then drive through their appropriate planets, carriers, and annuli, to the first speed annulus, so that the whole unit rotates together. Only one-third of the engine torque is reacted by the third speed sun wheel and clutch. As a consequence, the clutch needs to be only of relatively small dimensions.

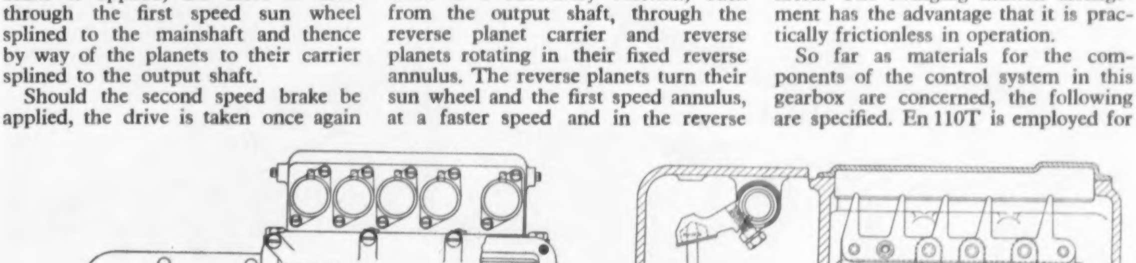
Reverse is obtained by applying a brake to the reverse gear annulus. The drive then passes, as in the case of the forward speeds, through the first speed sun wheel, planets and planet carrier to the output shaft. In this case, however, the output shaft is turned in the reverse direction. This is because there is a subsidiary reaction, back from the output shaft, through the reverse planet carrier and reverse planets rotating in their fixed reverse annulus. The reverse planets turn their sun wheel and the first speed annulus, at a faster speed and in the reverse direction, relative to the first speed sun wheel. This results in the first speed planets and their carrier moving in the reverse direction relative to the mainshaft.

Rear extension and control system. A ring nut on the ⅞ in diameter threaded rear end of the output shaft is tightened against the internally splined companion flange for the universal joint. In front of the boss of this flange is a thrower ring, a distance washer, a distance tube and the keyed on, En 32 speedo gear which is pulled against a collar round the shaft, immediately behind the transverse web. The rear ball bearing is mounted on the shouldered front end of the boss of the companion flange. Its outer race is clamped in a housing in the rear end of the gearbox by the bolted on rear cover, which houses a Gits oil seal.

Most of the control mechanism, including the busbar, ties and struts of the brake band and clutch operating mechanisms, is mounted on the bottom cover, which is bolted up directly to the transverse web and side and end walls of the box. It functions in a similar manner to that of most other Wilson type units. Two springs are employed in the spring gear, the outer one having a rate of 500 lb/in and the inner one 100 lb/in. The last time a full description of such an arrangement appeared in *Automobile Engineer* was March 1952.

The method of ensuring that two gears cannot be engaged simultaneously is somewhat unusual. Four components, termed by the manufacturers *swinging muttons*, are pivot mounted, so that they may swing in a vertical longitudinal plane, in a row immediately outboard of the struts that actuate the brake band toggles and the top speed clutch. On each strut is a knife edge which extends outwards below the gap between the muttons. When a gear is selected and the strut is lifted by the busbar, the knife edge is lifted into the gap and causes the muttons on each side of it to swing out of its way. This closes the gaps between the other muttons so that no other strut can be lifted. The swinging mutton arrangement has the advantage that it is practically frictionless in operation.

So far as materials for the components of the control system in this gearbox are concerned, the following are specified. En 110T is employed for



Plan and longitudinal section of the gearbox showing some of the preselector mechanism

the control cross shaft, and brake band and clutch actuating struts and tie rods. En 32 is used for the toggle levers and their support struts as well as for the spring carrier assembly. The busbar and its spindle are of En 8.

The electric preselector assembly is in a D.T.D.424 casting bolted to the left-hand side of the gearbox. Five solenoids are employed, one for each speed, and they are carried vertically in housings bolted on top of this casting. Screwed into the lower end of the solenoid core piece is a fork-end which is pinned to the horizontal arm of an En 8, 40-ton carbon steel, bell crank lever, pivoted on an En 8, 35-40 ton carbon steel spindle. The second arm of this lever is of such a length as to give a ratio of 2·2 : 1. It extends vertically downwards and, when the solenoid is energized, its end bears against a silver steel rod carried horizontally in a transverse plane, by two bosses cast in the base of the housing. The other end of this push rod is in line with the strut of the preselector mechanism.

When the solenoid is energized by operating the selector switch and depressing the pedal, the core moves approximately 0·2 in and causes the bell crank lever to force the push rod against the strut appropriate to the speed selected and to move it over the busbar. Then, as the pedal is released, the strut is lifted by the busbar and, acting through the tie rod and toggle lever mechanism, tightens the brake band on its drum. Self-adjustment for wear is effected by the usual device. The torque reactions required for the different speeds are matched with the braking force by using different toggle lever ratios for each brake.

Back axle

A Hardy Spicer, two-piece, open propeller shaft transmits the drive to the back axle. When the pre-selector gearbox is installed, the front part of the shaft is 19·51 in long, as measured between the outer faces of the universal joint flanges, and with the synchromesh unit it is 21·25 in long. In all installations, the rear portion of this shaft is 40 $\frac{1}{16}$ in in length. The outside diameter of each of the shafts is 2 in. An intermediate ball bearing is incorporated, and is pre-packed with grease. Its housing is carried on two large rubber bushes on the centre component of the cruciform bracing.

The axle illustrated is the Salisbury 2HA unit, but on later models, in the interests of rationalization so far as axle production is concerned, the 4HA unit is fitted. These axles are similar except that the offset of the pinion axis below the crown wheel axis and to one side of the plane of the differential spindle is different, and so is the pinion shaft bearing spacing, the crown wheel diameter and the spacing of the bearings supporting the differential cage. In both axles, the final drive ratio is 4·091 : 1. They are of the semi-floating type, with hypoid bevel gears, and a three piece casing. Both units are installed at a nose-up angle of 1 deg. Two and a half pints of extreme-pressure hypoid oil is specified. Under conditions where temperatures are generally higher than 10 deg F, S.A.E. 90 is recommended, but below this temperature, S.A.E.80 should be used.

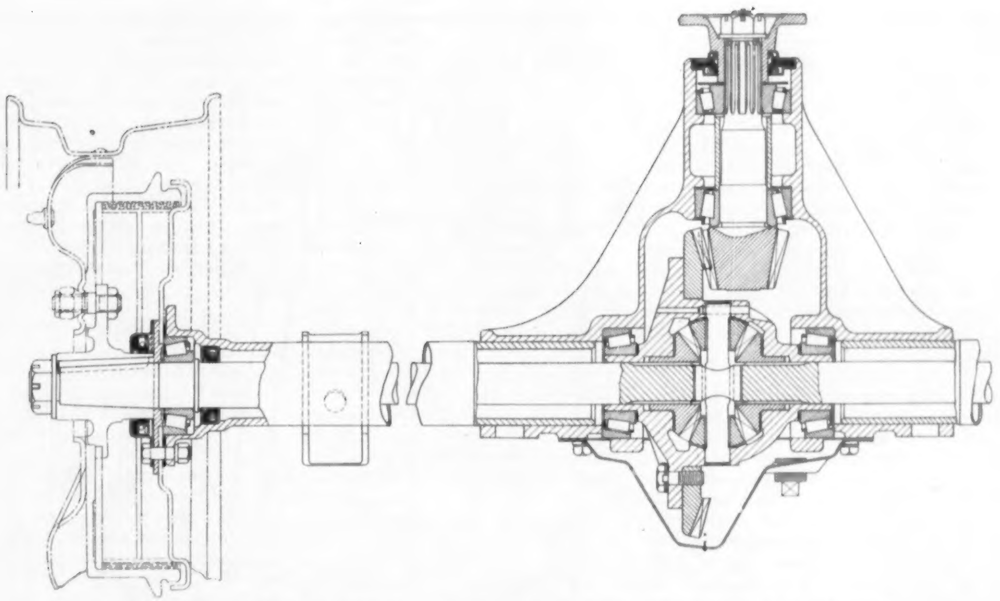
A Black Heart, malleable cast iron gear carrier is employed and the two axle tubes are pressed in and plug-welded. The bolted-on, pressed steel, rear cover carries a boss for the filler plug, and the oil drain plug is screwed into the base of the gear carrier casting. To stiffen the unit, gussets are incorporated between the bosses for the axle tubes and the nose of the casting.

The hypoid pinion is integral with its En 35A shaft and overhung from two taper roller bearings. In the 2HA unit, the axis of this shaft is offset 1·75 in below that of the crown wheel and 0·75 in from the plane of the differential pinion spindles, while in the 4HA axle, the offsets are 1·5 in and 1 in respectively. The shaft diameter at the rear bearing is 1 $\frac{3}{8}$ in, while at the front end it is 1 $\frac{1}{8}$ in diameter and splined to receive the companion flange for the universal joint. This flange is pulled on by a slotted nut on the $\frac{3}{4}$ in diameter threaded end of the shaft. A lip-type oil seal, supplied by Super Oil Seals and Gaskets Ltd., is housed in the end of the nose piece and bears on the boss of the companion flange. The inner periphery of the front end-plate of this seal assembly is turned forwards through 90 deg to form a lip which is shrouded by a U-section ring pressed on to the boss.

Clamped between the boss and the pinion are an oil thrower ring and the two taper roller bearings separated, in the 2HA unit, by a tubular distance piece. In the 4HA axle the inner race of the front bearing is located against a shoulder on the pinion shaft. The bearings are assembled one from the front and the other from the rear of the nose piece and their outer races bear against shoulders in their housings. In the 2HA unit, the spacing between the bearing centres is about 3 $\frac{1}{8}$ in and in the 4HA axle it is approximately 3 $\frac{3}{8}$ in. The axial position of the pinion is adjusted by means of shims between the outer race of the rear bearing and the shoulder in its housing, while the bearing pre-load of 8-12 lb-in is adjusted by means of shims immediately behind the inner race of the front bearing. Lubrication is effected by means of a passage cored in the nose piece, and leading from a point above the pinion into a space between the two bearings.

In the 2HA axle, the En 35A crown wheel is 9 $\frac{1}{4}$ in pitch circle diameter, while in the 4HA model it is 8 $\frac{1}{2}$ in pitch circle diameter. In both, 45 teeth are incorporated. The crown wheel is spigoted on to the flanged, one-piece differential carrier, which is of Black Heart malleable cast iron, and is pulled against the inner face of the flange by ten $\frac{3}{8}$ in diameter set bolts. Between the

The twin carburettor version of the 3·4 litre engine is mounted in this chassis



A semi-floating axle with a hypoid pinion and a three-piece casing are employed in the Sapphire

crown wheel and pinion, the backlash is 0·006-0·009 in.

Two En 35A differential pinions are carried on a $\frac{3}{4}$ in diameter spindle which is pressed into the carrier and positively located by a $\frac{1}{4}$ in diameter peg at one end. Spherical thrust washers, of En 2 and about $1\frac{3}{4}$ in outside diameter, are interposed between the pinions and the cage. The length of the tooth engagement between the differential pinions and wheels is approximately $\frac{5}{8}$ in. The wheels are about $3\frac{1}{4}$ in diameter and their bosses are $1\frac{5}{8}$ in diameter by $\frac{13}{16}$ in long. Flat thrust washers, of En 2, $2\frac{7}{8}$ in diameter are fitted between the differential wheels and the cage.

Two taper roller bearings housed in the gear carrier, with their centres spaced approximately 6 in apart in the 2HA unit, and $6\frac{1}{4}$ in apart in the 4HA, support the differential cage. The preload of the bearings, which is effected by springing open the carrier before assembly, is adjusted by shims between the inner races of the bearings and the cage. These shims are also used to adjust the mesh of the crown wheel and pinion.

Nineteen involute splines with a pitch circle diameter of 1·1875 in are hobbed on the inner end of each En 19 half shaft. Between the adjacent ends of the two shafts is an En 19 thrust block with an elongated hole in its centre, through which is passed the differential spindle. The half shafts are each $1\frac{3}{8}$ in diameter at the bearing and oil seal, and taper to $1\frac{5}{32}$ in diameter adjacent to the splines. This taper is incorporated because the bending moment on each shaft increases linearly from the inner end to the bearing. At the outer ends of the shafts, the En 8 wheel hub, or driving flange, is pulled on to a 1-in-12 taper, and the drive is furnished by a plain key. The brake drum is spigoted on to the driving flange in the conventional manner and secured by countersunk set screws. Five $\frac{1}{2}$ in diameter studs carry the wheel, which is secured by conical seating nuts with self-locking inserts in their ends. A collar is formed approximately mid-way between the ends of each stud which is screwed into the driving flange. A plain nut is then fitted and tightened against the inner face of the driving flange.

The brake back plate assembly is secured, together with the hub seal housing and bearing retainer plate, by five $\frac{3}{8}$ in diameter bolts to a flange up-set on the end of the $2\frac{1}{2}$ in diameter by $\frac{3}{16}$ in thick axle tube, which is of 0·30-0·35 per cent carbon steel. This tube houses the taper roller, wheel-bearing. The outer race is held in its housing by the retainer plate, on which is spigoted the brake back-plate, and the inner race is pulled against a collar machined on the half shaft.

Interposed between the back-plate and the carrier flange are shims that control the end float on the bearing, which is 0·006-0·008 in. Two grease seals, supplied by Retainers Ltd., are employed. One is housed inside the axle tube and bears on the half shaft immediately inboard of the collar that locates the inner race of the bearing. The other bears on the boss of the wheel driving flange and is carried in the housing, already mentioned. Between the two is formed the grease chamber for the bearing, which is served by a grease nipple on the axle tube.

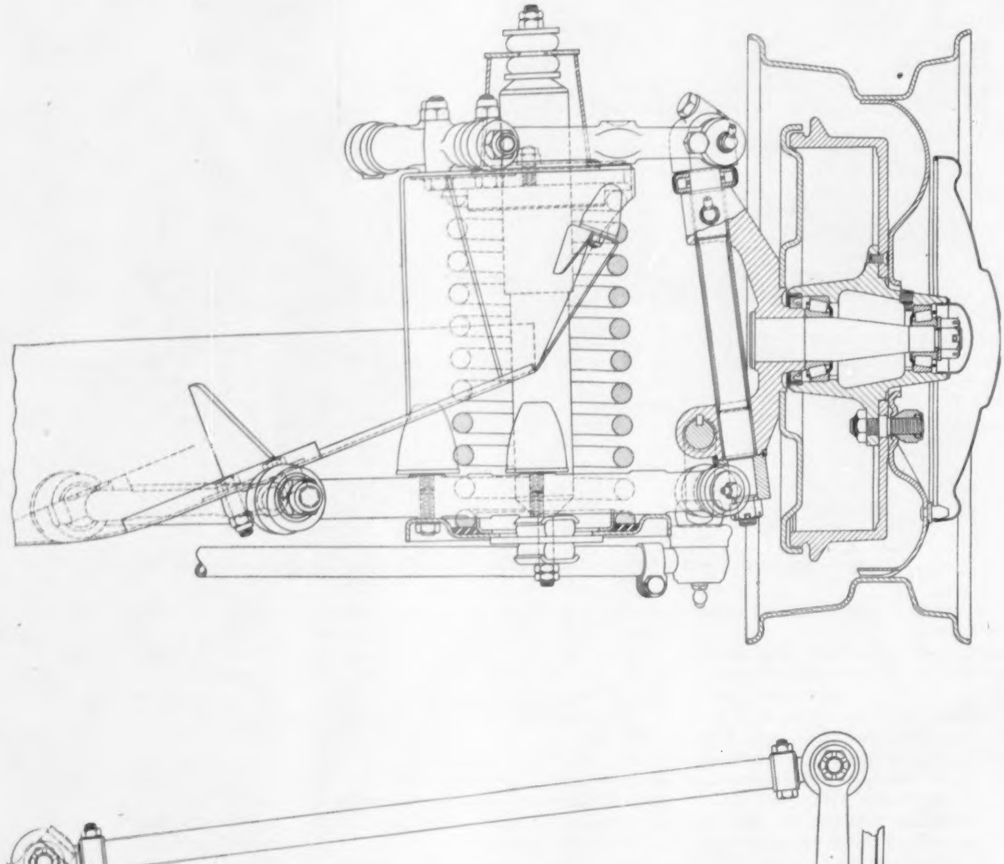
Rear suspension

A conventional rear suspension layout, with semi-elliptic springs and anti-roll bar, has been adopted. Damping is effected by Girling DAS/32 NF, telescopic shock absorbers, with a rebound setting of 200-230, and a bump setting of 100-130. The unsprung weight is 176 lb per side and the rear end rate is 109 lb/in. To the laden position, the deflection is 8·75 in with a load of 950 lb per side. This gives a periodicity of 74 oscillations per minute. Deflection to full bump is 12·75 in, the free camber is 7-7·5 in, and the loaded camber is 1·25-1·5 in.

The springs, which are each $50\frac{1}{2}$ in long and which are slung below the axle, are supplied by R. Berry and Sons. Ten leaves, 2 in wide, are employed, and they are shot peened to improve their fatigue resistance. The top two are $\frac{7}{32}$ in thick and the remainder are $\frac{3}{16}$ in thick.

The axle is offset $2\frac{1}{4}$ in forward of the spring centre to give a slight under-steer tendency, and an additional under-steer effect is induced by the fact that the front spring eye is $4\frac{1}{2}$ in below the rear one. Either a Silentbloc or a Metalastik bush is specified for the front eye, while for the rear, a Silentbloc-Harris CP shackle assembly is fitted. At the front, $\frac{5}{8}$ in diameter pins are employed and they are carried between plates bolted one on each side of each frame side member. The upper pin of the shackle assembly is carried in a tube welded in the frame side member. At full bump the axle comes against Metalastik, rubber bump-stops on the underside of the frame side members. The rebound stops are incorporated in the shock absorbers.

An En 45 anti-roll bar is employed. Its diameter and effective length are $\frac{11}{16}$ in and $22\frac{3}{4}$ in respectively, and the effective length of the radius arm is $9\frac{1}{2}$ in. The bar is carried, forward of the axle, in two rubber bushes in steel clips bolted one under each side member of the frame, and it is attached by means of a drop link on each side to one end of a $\frac{3}{16}$ in thick plate welded



ARRANGEMENT OF THE FRONT SUSPENSION

The shock absorbers may be changed without removing any other suspension components

to the spring clamping plate. A rubber bushed eye at the top end of the drop link carries the pin by means of which it is secured to the bar, and at the bottom end, two spherical faced rubber washers seat one on each side of the anchorage plate. This plate extends under the axle to the rear where it carries the rubber sandwich type fitting at the lower end of the shock absorber. The upper end-fitting on the shock absorber is a rubber bushed eye and it is overhung-mounted on a bolt in the frame cross member.

Front suspension

A double transverse-wishbone-link type front suspension system, with coil springs, has been adopted. The links trail at an angle of 25 deg to enable the front cross member to be shaped so as to pass under the front end of the sump. When the wheels are in the straight ahead position, the swivel pin angle is 5½ deg and the camber angle 2 deg, while on 25 deg lock, the angles are 6 deg and 1½ deg respectively, as viewed in a direction parallel to the plane of the wheel. When the wheel is in the straight ahead position, the camber angle change from the fully laden to the full bump positions is 1 deg, and the track change is 0·3 in. In the laden condition, the castor angle is 1 deg, while unladen it is zero. The toe-in is $\frac{3}{16}$ in.

The spring rate is 184 lb/in and this gives at each wheel a rate of 89 lb/in and a periodicity of 74 oscillations per minute. To the fully laden position the spring deflection is 6·5 in and to full bump it is 8·5 in. A total of 10½ coils is incorporated; their overall diameter is 5¼ in and the wire is 0·585 in diameter. The unsprung weight is 114 lb per side. Both ends of each spring seat on rubber rings approximately ¼ in thick and they are radially located by circular, cupped pressings welded to the spring pans.

Girling, DAS 6/47 NF telescopic shock absorbers are mounted co-axially with the spring. Sandwich-type rubber-mountings are employed at each end. The lower rubbers are rectangular in shape, except that their seating faces are of semi-circular section to accommodate angular deflection in one plane. They are positioned one each side of a plate bolted to the spring pan which in turn is bolted up to the lower wishbone arms. At the upper end, the rubber washers are circular in shape and are carried on each side of the top plate of a fabricated turret, secured by three bolts to the spring pan. The shock absorber settings are 120-125 for the rebound and 100-120 for the bump strokes. Metalastik bump and rebound stops are employed. The two rebound stops are mounted on brackets, one in front of and the other behind the spring pan, where they are struck by the upper wishbone arms. Two bump stops are also fitted; they are mounted one on each lower wishbone arm, and strike the lower flanges of the spring pan assembly. The flanges are each supported above the point of contact by a gusset extending up the side of the spring pan to a point beneath the pivot of the upper wishbone link.

Another anti-roll bar, in addition to that used at the rear, is employed at the front. This bar is also made of En 45, silicon manganese spring steel. Its diameter and effective length are $\frac{11}{16}$ in and 31⅞ in respectively, and the effective length of its radius arms is 7½ in. The bar is carried by two rubber bushes secured under the frame side members by means of steel clips. The drop link from each end of the bar is attached to a bracket bolted on the front arm of the lower wishbone link. Spherical faced rubber washers are fitted at both ends of this link.

Both wishbones are two-piece, I-section, En 8 stampings. The upper one is 8⅜ in long between bearing centres, while the lower one is 17 in long. As measured perpendicularly from the longitudinal centre-line of the chassis, the distance to the centre of the upper pivot pin is 16½ in while that to the centre of the lower pivot pin is 8¾ in. Between the centres of the upper bearings at each end of these pins, the spacing is 5⅜ in and it is 14⅞ in between the lower ones. The upper and lower pivot bearings are spaced vertically 9·9 in apart. As measured vertically, the distance between the upper and lower bearings at the outer ends of the arms is $10\frac{1}{16}$ in.

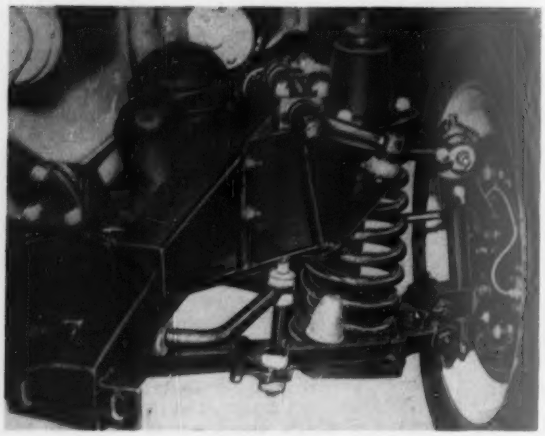
Metalastik bushes are employed in the relatively heavily laden but widely spaced pivots of the lower wishbone arm, while screwed and plain bushes are used in the more lightly loaded upper pivots. The Metalastik bushes are carried on $\frac{11}{16}$ in diameter pins formed at the ends of an En 12 pivot forging. This forging has four lugs formed on it, two adjacent to each bearing, by means of which it is bolted up to brackets on the front and rear walls of the frame cross member. On each end of the pins, a self-locking nut pulls the centre tube of the bush against a distance piece shaped to clear the fillet radius between the pin and the centre portion of the forging.

Between the outer ends of the lower wishbone arms, the En 8 swivel pin bearing boss is carried on a screw type bearing. The screw bush, which is pressed into the boss, is of En 6, as also is the pin, which in addition is Parkerized. A flange round the rear end of the bush takes the axial thrust from the swivel pin bearing boss. Interposed between the ends of the bush and the wishbone arms are rubber sealing rings.

On assembly, the pin is passed through the 0·8125 in diameter hole in the end of the front wishbone arm. Then the first rubber sealing ring is placed in position and the bush and boss assembly is threaded on. Next, the second sealing ring is passed over the pin, and finally, the steel bushed rear wishbone arm is assembled on to the 0·5625 in diameter, rear end of the pin and secured by a self-locking nut. The reason for bushing the hole in the end of the rear arm of the wishbone is to make it common with the front arm. Lubrication of the ¾ in diameter screw type bearing is effected by means of a nipple screwed into the head end of the pin. The lubricant passes along an axial passage and out through three radial holes to the bearing surface.

An En 8 forging carries the pivot at the inner end of the upper wishbone link. It is secured to the top of the spring pan assembly by means of three ⅜ in diameter bolts, two adjacent to the bearing at one end and one at the other, but instead of having the pins formed at its outer ends, as is the case with the lower pivot bearing, it is hollow, and carries a separate, Parkerized En 6 spindle in two bearings inside it. The wishbone arms are assembled on to the ⅝ in diameter shouldered ends of this spindle and secured by self-locking nuts. Axial thrust is taken at the front bearing, which is of the threaded type. The thread is cut directly in the hollow forging, and the root diameter of the threads on the pin is 0·73 in. On the other hand, the rear bearing is a plain, Clevite bush and therefore does not

A slight understeer tendency is obtained by mounting the axle forward of the spring centre and by inclining the spring upwards from front to rear



A gusset supports the flange that is struck by the rubber bump stop

of which are retained in the bearing bosses.

The steering arm is pulled into a tapered hole in the lower boss by a slotted nut on its $\frac{5}{8}$ in diameter threaded end. It is positively located by means of a plain key. The boss that carries the outer end of the bottom wishbone link is mounted on the $\frac{3}{4}$ in diameter lower end of the swivel pin and is pulled against the boss for the lower swivel pin bearing by a slotted nut on the $\frac{1}{2}$ in diameter threaded end of the pin. A rubber sealing ring is housed in a groove machined partly in the wishbone bearing boss and partly in the swivel pin bearing boss.

have to be positively located axially relative to the front one. Its inside diameter is $\frac{3}{4}$ in. Rubber sealing rings are interposed between each end of the hollow forging and the wishbone arms. Lubrication is effected by means of a nipple, screwed in near the centre of the forging, through which the lubricant is passed into the space round the waisted portion of the pin, whence it passes to the bushes at each end.

At the outer end of the upper wishbone link, there is another threaded bearing, but the thread is cut in an En 6 bush clamped in the split boss integral with the top end of the swivel pin. Its clamping bolt registers in a groove round the outer periphery of the bush to provide positive axial location. The thread on the pin is Parkerized and its root diameter is 0·73 in. Lubrication is effected, through a nipple, in a similar manner to that of the lower bearing. Formed on the pin, which is of En 6, and which is free to rotate in $\frac{19}{32}$ in diameter holes in the bosses in which it is carried in the wishbone arms, is a hexagon-shaped collar adjacent to the front face of the rear arm of the wishbone. Thus, adjustment to the castor angle can be made by using a spanner on the hexagon collar, and the setting is fixed by means of a self-locking nut which clamps the rear wishbone arm against the hexagon collar.

An En 16T swivel pin is employed and it is Parkerized. Integral with its upper end is the split boss for the outer bearing of the top wishbone. Interposed between this boss and the upper boss of the En 12 forging that carries the stub axle is a Timken T88 shrouded roller thrust bearing. The two Clevite, lead bronze lined, steel bushes for the swivel pin bearings are pressed into their bosses on the stub axle carrier forging, and their centres are spaced 6$\frac{3}{8}$ in apart. Both bearings are 1·3 in long, the inside diameter of the upper one is 0·88 in, while that of the lower one is 0·87 in. Between the two bearings, the pin is waisted to 0·8 in diameter and surrounded by a 1 in diameter by 16 S.W.G. tube, the ends

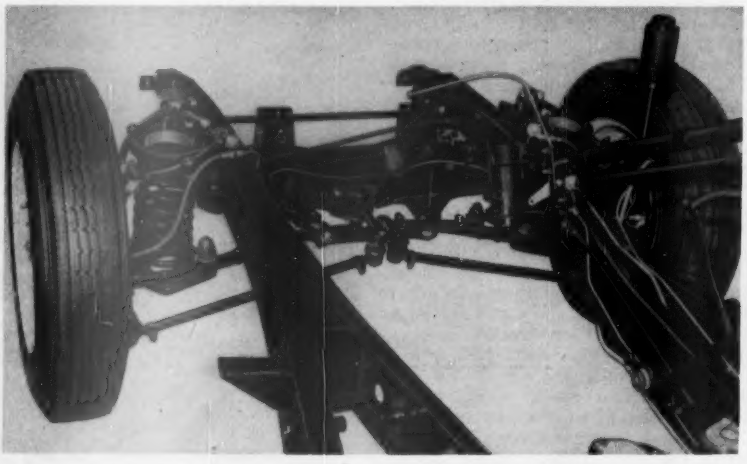
A head is formed on the inner end of the En 16T stub axle which is pressed into a 1$\frac{1}{4}$ in diameter hole in its carrier forging. The axle is 1$\frac{1}{4}$ in diameter at the inner wheel bearing and $\frac{3}{4}$ in diameter at the outer one. Carried in the malleable cast iron hub are the two taper roller bearings, which are assembled one from each end, and a Gaco MIS 113 oil seal housed in the inner end. This seal bears on the periphery of the boss of the stub axle carrier forging. A slotted nut and plain washer on the $\frac{5}{8}$ in diameter threaded end of the stub axle pulls on the hub assembly. An end float of 0·004–0·006 in is permitted in the bearings. The outer end of the hub is closed in the usual manner by a pressed steel cap. Lubrication during service may be effected by removing a screw plug from a hole in the hub through which grease may be inserted into the space between the two bearings.

The cast iron brake drum is spigoted on to the hub flange to which it is secured by countersunk set screws. Five $\frac{1}{2}$ in diameter studs are each passed through a hole in the brake drum and screwed into the hub flange. A collar, machined approximately midway between the ends of the stud, registers in the hole in the brake drum and is tightened against the flange. The whole assembly is locked by means of a nut tightened against the inner face of the flange and welded to the stud. Conical-seating nuts with self-locking inserts in their outer ends secure the wheel. The brake back plate is spigoted on to the stub axle carrier forging and secured by four $\frac{3}{8}$ in diameter bolts and nuts.

Steering

Burman recirculatory - ball type steering gear is employed. The steering box is about 6$\frac{1}{2}$ in behind the front wheel axes and it is mounted on the frame side member. In the straight ahead position, the ratio is 22:1. This gives three turns of the 18 in diameter steering wheel from lock to lock. The angle of the inner wheel in the full lock position is 31 deg while that of the outer wheel is 24$\frac{1}{2}$ deg. This gives a turning circle of 42 ft 6 in. The steering wheel is adjustable, its telescopic movement being 3 in.

A two-piece track rod layout has been adopted. The drop arm, which is an En 12 stamping with an effective length of 6 in, is connected by means of a tubular rod, 0·865 in outside diameter by 0·615 in inside diameter, to the slave lever pivoted between lugs on the rear face of the front cross member. Alford and Alder self-adjusting ball joints are screwed into the split ends of the tube and the setting is locked by clamping rings tightened by $\frac{5}{16}$ in diameter bolts and nuts. The thread on the ball joint at one end of the tube is right-handed and at the other end it is left-handed, so adjustment may be effected simply by rotating the tube after loosening the



The front cross member, on which are assembled the steering and suspension components, is bolted solidly to the frame

clamping rings. The centre-to-centre length of this rod is about 9 in.

The En 12 idler lever is carried on a $\frac{3}{4}$ in diameter, En 32 pivot pin, the ends of which bear in two Clevite bushes. These bushes are spaced with their centres 2·6 in apart and they are pressed into bosses, welded one into each of the lugs of the bracket on the frame cross member. The length of the upper bush is 1·05 in and that of the lower one is 1·70 in. On each side of the axis of the slave lever and about $7\frac{1}{4}$ in, as measured along it, from the axis of the pivot pin, are the inner end-fittings of the two-piece track rod, with their centres $2\frac{1}{8}$ in apart. The centre of the steering rod end fitting on the slave lever is about 6 in from the pivot pin axis.

Each of the two pieces of the track rod has an effective length of 23 in, and the cross sectional dimensions of the tubes are 0·865 in outside diameter and 0·615 in inside diameter. All the end fittings and their method of attachment to the rods are the same as described for the steering rod. The En 12 steering arm has an effective length of $5\frac{3}{8}$ in.

Brakes

Girling hydraulic brakes are employed. Two leading shoe, HLS/S, units are incorporated at the front and two trailing shoe, HNS/S/H, at the rear. The 11 in diameter drums are of Millenite cast iron. At the front, the shoe area is 93 in^2, while at the rear it is 91 in^2 and in all brakes, the shoe width is $2\frac{1}{4}$ in.

The brake pedal is carried on a perforated brass strip bush, 2 in long, on a $\frac{3}{4}$ in diameter En 6 pivot pin in the angle between the cruciform and the frame side member. A lever ratio of 5·4:1 has been adopted, and the pedal travel is 6 in. The lower end of the pedal is pinned directly to the piston rod of the master cylinder. A pistol grip hand brake control is employed and the cable from its end is attached to a relay lever, the ratio of which is 10:1, pivoted under the centre box-member of the cruciform bracing.

A short adjustable tie rod attached to the lever is pinned to the centre of a pressed steel stirrup round which is passed the cable, the ends of which are taken through conduits and attached one to each rear brake. The front ends of the conduits are carried between lugs on the rearmost of the two tubular, intermediate cross members, and their rear ends are secured to brackets on the brake back plates.

Frame

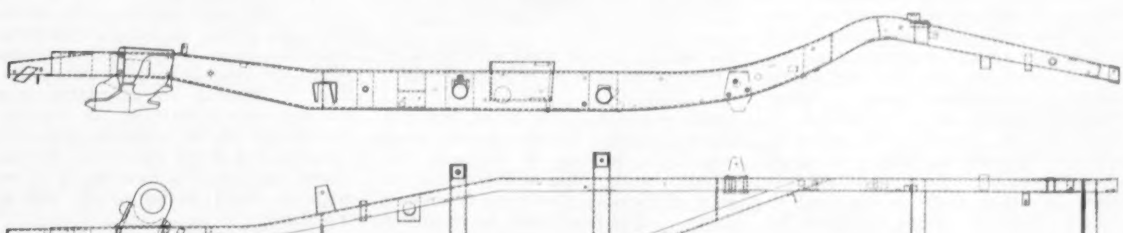
A simple frame structure comprising two side members and five cross members, together with a form of cruciform bracing, is employed. Except where otherwise stated in this description, the main members are of 14 S.W.G., 28–32 ton carbon steel. At the front, the width over the side members is 31 in, and the frame is swept outwards behind the front suspension so that at the centre and rear it is 46 in wide. The side members are swept upwards over the top of the back axle and then down again to a suitable level to take the bumper irons at the rear.

These side members are of channel section and their depth is $3\frac{3}{8}$ in at the front, 6 in at the centre and 2·05 in at the rear. Each is closed by another channel section extending from a point at the front face of the front cross member for a distance of approximately 18 in to the rear where the side member is turned outwards. The closing member is then swept inwards and extended back to form the front portion of the cruciform bracing. At the centre of the frame, it is swept outwards again to form the rear portion of the cruciform, and its rear end is welded inside the channel section of the side member, approximately 12 in behind the axis of the spring eye. The centre portion of each of the two channel section cruciform members, that is from the front of the first of the two tubular intermediate cross members to the back of the second one, is boxed-in by a closing plate welded between the flanges. At the centre, the inner faces of the cruciform members on each side are 14 in apart, and they are joined by a box-section bracing member.

The rear cross member is an open channel section, 2 in deep by 2 in over the flanges. Above and slightly to the rear of the axle is another cross member, also of channel section, but 4 in deep by 2 in over the flanges. The ends of this channel are boxed in by means of a closing plate welded to the flanges and they carry the tubes for the trunnion pins on which the upper ends of the telescopic shock absorbers are mounted. The two intermediate cross members are, as has already been mentioned, of tubular form; they are 2 in diameter by 14 S.W.G. mild steel. Closing plates, 5 in long, are welded to the flanges of the side members where these two cross members are passed through the assembly to form body mounting brackets overhung from each side.

The front cross member is trapezoidal in section, and it is cranked to pass under the sump. Its cross sectional dimensions are $5\frac{1}{2}$ in wide, $4\frac{1}{2}$ in deep at the front and 3 in deep at the rear. The ends are swept backwards so that its centre portion passes under the front of the sump. This member is secured to two brackets, one on the inner face of each side member, by six $\frac{3}{8}$ in diameter bolts. Four more bolts of the same diameter are passed laterally through each side member and the flanges of the 12 S.W.G. spring pan brackets, which are welded to the ends of the cross member.

At the rear, two $\frac{1}{4}$ in thick, steel plates are secured by three $\frac{3}{8}$ in diameter bolts to each side frame to carry the front eye of the spring. Seven body mounting points are incorporated on each side of the frame. The front point and numbers 6 and 7, are 0·08 in thick steel plates welded underneath the



The main members of the frame are fabricated from 14 s.w.g. channel section, the side members being boxed-in at the front and rear

top flange of the side member and tapped $\frac{3}{8}$ in B.S.F. Number 2 point is an inverted channel section bracket, 2 in wide by 3 in deep, drilled for a $\frac{3}{8}$ in diameter B.S.F. bolt, and number 3 point is a steel plate welded on top of the first of the two tubular cross members, also tapped $\frac{3}{8}$ in B.S.F., but incorporating a locating dowel. The fourth one is a similar plate, but without the dowel, welded on the second cross tube. An outwardly turned flange on the outer plate that carries the spring eye forms the fifth body mounting point.

Electrical equipment and other features

All the electrical equipment is of Lucas manufacture. A GWT 11A battery of 12 volts output and a capacity of 24 amp-hr is employed. It is charged by a C45 PV5 dynamo used in conjunction with an RB106/1 voltage regulator. An SF6 fuse box protects the accessories while an SF5 fuse box is incorporated for the preselector circuit. The head lamps are PF770 units, 490-type side lamps are employed, and the tail, stop and reversing lamps are of the 464 type. Other electrical equipment installed includes the WT614 dual tone horns and SF80 trafficators.

There are twenty-seven lubrication points on this chassis. Twenty-one of these are oil nipples, two are grease nipples, two more are holes, sealed by screw-in plugs, through which grease is inserted, and there are two screw-down greaser caps. So far as jacking is concerned, Smith's Bevelift equipment is provided. To accommodate the jack, two square section tubes are welded one to each side member of the frame at a point just to the rear of the back wheels. At the front, two more square-section tubes are welded one to the end of each frame side member. The spare wheel is housed in a separate compartment under the luggage platform in the boot.

Two $1\frac{3}{4}$ in diameter by 14 S.W.G. down-pipes carry the exhaust gases from the manifolds to an oval-section, Burgess expansion box. This box measures 5 in by 3 in by 11 in long. It is connected, by a single $1\frac{3}{4}$ in diameter 18 S.W.G. pipe, to a 5 in diameter by 24 in long, Burgess silencer. The tail pipe is again $1\frac{3}{4}$ in diameter by 18 S.W.G. The system is mounted on three metal clips, the upper ends of which are rolled over to form eyes in which rubber bushes are carried to insulate the frame from vibration.

ASHANCO EXHAUST BRAKE

An Electrically Controlled Unit for Diesel-engined Vehicles

IN some parts of the world, particularly in mountainous countries such as Switzerland, exhaust brakes have been popular for many years. The principle of operation of these units is well known: they all consist essentially of a valve unit by means of which, when the brake is applied, the exhaust pipe is closed. Under these conditions, the braking force is supplied by the engine acting as a compressor and pumping air into the closed exhaust system. The effectiveness of the unit may be increased, of course, by engaging a low gear.

There are a number of advantages to be gained from installing a brake of this type for use in conjunction with a conventional braking system. It gives additional safety, in that it may be used in the event of failure of the mechanical or hydraulic system, since it is completely independent of the other brakes. Exhaust brakes are not subject to fade, as are shoe-and-drum type units. They add to the overall braking efficiency and, if used to the best advantage, reduce brake-lining wear. Other benefits claimed are: a reduction of driver fatigue, because little effort is required to apply the brake; reduction in engine wear since, when an exhaust brake is fitted, there is no need to change down to such a low gear for descending hills; an increased tyre life as a result of more even braking as well as lower brake drum and wheel temperatures. There is also less danger of the engine over-running the governed speed when the vehicle is descending hills.

A new unit, termed the Ashanco exhaust brake, has recently been introduced by Thomas Ash and Co. Ltd., of Birmingham, 5, for application to diesel-engined chassis. This unit has a number of advantages in addition to those already mentioned. It is electrically operated and, therefore, control is exceptionally light. Control is effected by means of a switch unit mounted on top of the pedal for the conventional brakes, so the driver can apply it simply by resting his foot on the pedal.

This means that not only does the exhaust brake come into operation

The Ashanco exhaust brake with the cover removed to show its actuating linkage

before the normal foot brake, but it is easier to apply. For this reason, when only light braking is required, the exhaust brake alone will probably be used. With most other types of exhaust brake, control is effected by means of a separate, hand-operated lever so there may be a tendency for the driver to ignore it and to use his foot brake in the normal manner.

Another advantage of the Ashanco brake is that there is little likelihood of the driver depressing the accelerator with the brake in the on position, whereas with a manual control, this is a possible contingency. The electrical control shows up to advantage when underfloor engines, positioned some distance from the driving position, are employed. This is because in these installations, a mechanically operated exhaust brake requires a relatively complex system of control rods and levers with numerous pivot joints, all of which are subject to wear. Moreover, such systems may be adversely affected by frame distortion.

The body of the Ashanco exhaust brake is a malleable cast iron sleeve, in which is mounted a butterfly valve. The flanged ends of the sleeve are conical-faced and project about $\frac{1}{16}$ in to seat in mating flanges on the engine exhaust pipe and the pipe to the silencer. Cast integrally with the unit is a platform, which is offset to one side of the sleeve, and on top of which is mounted the solenoid. The core of the solenoid is passed through a hole in the platform and connected by means of a fork joint to a lever, the centre of which is mounted on a spindle carrying the butterfly valve operating in the sleeve. To the other end of this lever is attached a tension spring which, when the brake is not in operation, holds the valve in the open position. When the valve is in this position, the lever is against an adjustable stop screwed into a boss on the casting. Completion of the circuit energizes the solenoid and closes the valve. In the closed position, the butterfly is at an angle of about 35 deg to the axis of the sleeve.

Arrangements have been made for sufficient air to leak past the valve to allow the engine to idle normally when the exhaust brake is on and the vehicle brought to a standstill by the foot brake. All the operating mechanism is, of course, outside the sleeve, and is enclosed by a pressed steel cover secured by set screws to the main casting below the solenoid platform. Gas-tightness of the valve spindle bearings is ensured by fitting copper and asbestos washers at their outer ends. Either 12- or 24-volt electrical equipment can be supplied.

FRONT SUSPENSION

Some Aspects of Divided-axle Systems

D. R. Hume

THE divided-axle, or single-transverse-link, suspension layout has two potential disadvantages and a number of practical advantages. The two apparent disadvantages are its relatively large track variation and the gyroscopic phenomenon known as precession. In many quarters, the gyroscopic effect is regarded as being of such a magnitude as to make such suspension systems unacceptable. However, it is intended to show that this is not so.

When a wheel spinning in a vertical or nearly vertical plane is tilted, it will instantly commence to rotate, or castor, if free to do so, in the same sense as the direction of tilt, about its vertical or near vertical axis, Fig. 1. This is known as precession. The precessional torque thus developed is given by the expression $T=(W/G)K^2\omega\omega_T$, where W is the weight of the wheel, K the radius of gyration, ω the rate of rotation of the wheel and ω_T the rate of tilt. Therefore, the variables that influence the magnitude of the precessional torque are the weight of the wheel, its rate of rotation which, of course, depends on the speed of the vehicle, and the rate of tilt. In a single-transverse-link suspension this rate of tilt is dependent upon the upward acceleration of the wheel swinging about the link pivot pin.

It would therefore appear that each time the wheels of a vehicle incorporating this type of suspension strike a bump they will, on rising, rotate about their swivel pins, and on returning to the road, be brought rapidly back into the straight ahead position by the aligning torque of the tyres. This castoring motion will at the same time be transmitted through the steering linkage and gearbox to the steering wheel. However, for the following reasons, the road wheels are not free to precess in this manner. First, when both wheels strike a bump simultaneously, the precessional tendencies oppose each other and consequently cancel out. The resultant loading of the track rod system under these conditions will not at any time be as great as that imposed on it by fast cornering on rough surfaces. On the other hand, when one wheel only traverses a bump large enough to give rise to appreciable wheel tilt, the precessional torque is largely opposed by the aligning torque and by the inertia of the other wheel, and the force transmitted to the steering wheel is usually very small. Also the inertia effect produced by the acceleration of the masses, and the aligning torque both vary with the speed of the vehicle, and the ratio of their magnitude to that of the precessional torque is more or less constant. Therefore, as the vehicle speed increases, the loads transmitted to the steering wheel remain very small. However, there is a critical speed at which the load transmitted may be appreciable, and the reasons for this phenomenon are as follows.

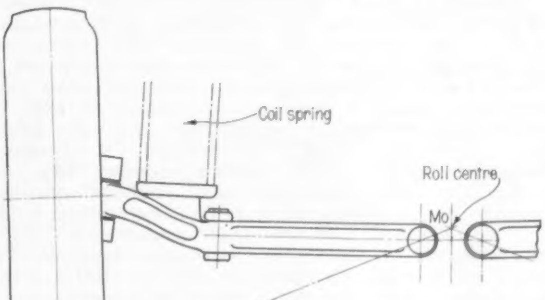


Fig. 2. A typical divided-axle front suspension layout

Maximum precessional torque on one wheel is obtained when it drops into a pothole to the full extent of the rebound travel and then, on striking the far edge, is accelerated upwards to the full extent of the bump travel. For any given relationship between spring rate, tyre pressure and damper setting, this can only happen when the road speed is such that the wheel will reach full rebound just as it strikes the far edge. Should the road speed be lower or the pothole of considerable length, the vehicle will tilt down on to the wheel in the pothole, thereby increasing the spring load and reducing the possible range of upward movement. If the road speed is higher, either the wheel will not drop to the full extent or if it does, as in a long pothole, the effect of aligning torque will be added to that of the inertia of the other wheel, and the two forces together will be great enough to prevent appreciable precessional deflection.

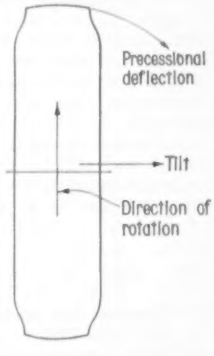


Fig. 1. Plan view of a tilting wheel

Tests have shown that the critical speed range is confined to narrow limits. For instance, a certain vehicle fitted with single-transverse-link front suspension, Fig. 2, with the tyre pressures set at 22 lb/in², was driven so that one wheel traversed a manhole cover some 2-3 in below the general road surface. Under these conditions, the critical road speed, at which movement due to precessional torque was noticeable at the steering wheel, was 26 m.p.h. The pressures were then raised to 25 lb/in², and the critical road speed at which maximum precession and consequently steering wheel deflection took place dropped to about 24 m.p.h. Moreover, for a given tyre pressure, the manhole cover could be traversed at some 2 m.p.h. faster or slower than the critical speed and no angular deflection of the road wheels, and consequently of the steering wheel, occurred.

In a certain vehicle, the sprung weight of which is about 1,470 lb, and which has 13 in diameter wheels and 6·40 in tyres weighing together approximately 37 lb per assembly, no gyroscopic effect is noticeable. On the other hand, if 15 in diameter wheels with 5·50 in tyres, weighing approximately 45 lb, are fitted to the same car, a gyroscopic effect of small magnitude may be felt. With a larger vehicle, the sprung weight of which is approximately 2,500 lb, and on which 16 in diameter wheels and 6·25 in tyres, together weighing about 56 lb, are used, the gyroscopic effect is sufficiently marked for it to be possible to define positively the conditions under which it takes place. In fact, it was on this vehicle that the figures given in the previous paragraph were obtained.

Practical experience has shown that track variation, even of considerable magnitude, is not necessarily detrimental unless there are inconsistencies in the track rod geometry that cause interference with the straight running of the wheels. Even then, only in extreme cases have any adverse effects been observed. On a single link arrangement with a ratio of R : L of about 1 : 2, where R is the tyre rolling radius and L is the distance between the link pivot and tyre contact centre, the track variation, for a given wheel deflection, can be smaller than that of a double parallel-wishbone-link arrangement in which both links are of normal proportions but of equal length. Moreover, the variation is no

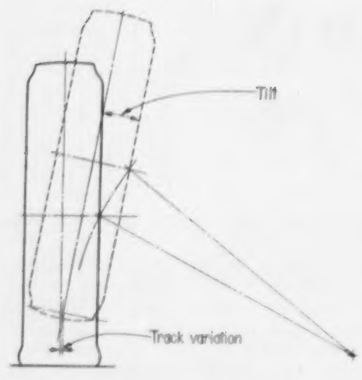


Fig. 3. Track variation with a low roll centre

greater than some unequal - length-wishbone arrangements. It is well known that, because of tyre flexibility, this track variation is accommodated by a rolling action and not by scrub; therefore it does not give rise to undue tyre wear.

If the pivot points of the single - link suspension arrangement were close to the ground, the track variation for a given deflection would be a minimum but wheel tilt would be large. Conversely, if the pivot points are level with the wheel centres, track variation, for a given wheel deflection, is large but wheel tilt is a minimum, Figs. 3 and 4. This arrangement is obviously preferable, but is rarely practicable because, in a car with the engine at the front, it is usually necessary to make a compromise between the conflicting requirements of the suspension geometry and the frame and engine and radiator installation height. With the normal track and wheel dimensions of present-day cars, the optimum pivot position is a distance of about 25 per cent of the tyre rolling radius below the level of the wheel centres. These proportions give approximately 1½ per cent track variation for 3 in deflection of one wheel upwards from the static position.

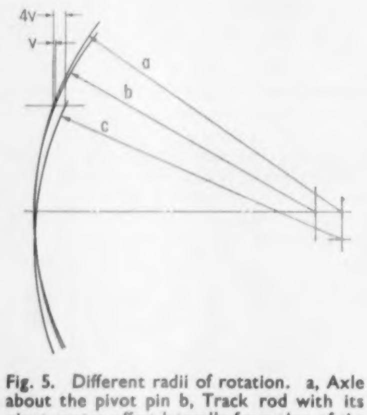


Fig. 5. Different radii of rotation. a, Axle about the pivot pin b, Track rod with its pivot centre offset laterally from that of the stub axle: c, Track rod with its pivot centre offset vertically from that of the stub axle

Single - transverse - link suspension systems have four main advantages. They are: simple and accurate track rod geometry, high roll centre, and consequently no need for an anti-roll device, and simplicity of construction. It is not difficult to obtain good track rod geometry with a single-transverse-link arrangement provided the axes of the pivots of the links and of the radius rods, which react brake torque and wheel alignment, are parallel with the longitudinal centre line of the vehicle. So far as the other aspects of geometry are concerned, some liberties can be taken without noticeable adverse effect, provided the centres of the ball-joints on the slave arm are at the same level as the axes of the link and radius arm pivots. This is fortunate because the bosses of the link and radius rod are considerably larger than the ball joints on the slave arm, so their centres must be spaced apart a greater distance than is necessary between those of the ball joints. To space the ball-joint centres the same distance apart as those of the transverse-link pivots would call for a heavy lever, and the twisting moment due to the angularity of the divided track rod, as viewed from the front, would be greater.

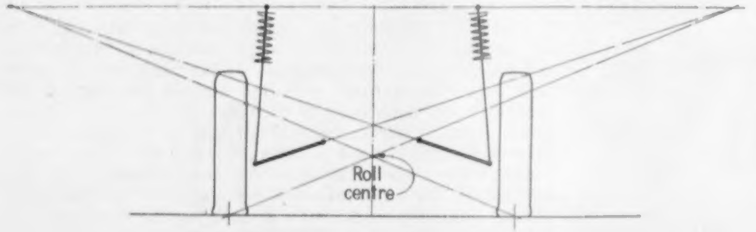


Fig. 6. A diagrammatic layout of the Ford front suspension system

It will be appreciated that the adverse effect produced by positioning the ball-joint centres above or below the level of the pivot axes of the suspension member is much greater than that resulting from moving the ball-joint centres an equal distance out of line laterally, Fig. 5. To avoid steering wheel deflection due to geometrical inconsistencies, it is necessary to employ a transverse drag-link between the steering box and the slave arm which must be mounted mid-way between the front wheels; in any case, this arrangement is usually the most convenient one. A short link, which is inherently stiff and light, may be used and this gives a positive feel to the steering.

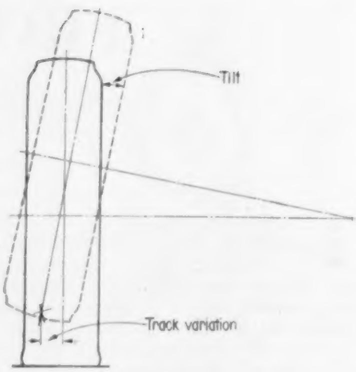


Fig. 4. Track variation with a high roll centre

Rack-and-pinion steering is not recommended for use with this type of suspension unless some damping device is incorporated. This device may take the form of fibre washers under the king pin heads, as on some double-transverse-link arrangements; or it may be a hydraulic shock absorber with leak characteristics giving a low torque setting. Another essential requirement for rack-and-pinion steering systems applied to this type of suspension is, of course, that the spacing and level of the centres should be correct. This is not practicable on a chassis in which the engine has been mounted as far forward as possible, that is, immediately behind, or partly overhanging, the front cross member unless the track rods are forward of the axle. However, with some hub, brake backplate and road wheel designs, it is not possible to arrange the track rods in this forward position. In general, the Ackerman principle should be followed as closely as possible not only in single-transverse-link suspension systems, but also in double-transverse-link and vertical-pillar arrangements, as well as with the beam axle. Again, this is sometimes difficult unless the hub assembly has been designed with this layout in view.

The vertical distance between the roll axis and the centre of gravity, and therefore the rolling moment, of a vehicle with a single-transverse-link suspension will be approximately 50 per cent less than that of the same vehicle with a double-transverse-link arrangement. Moreover, when this single-link layout is employed in conjunction with a conventional through-axle at the rear, or a De Dion layout incorporating a Panhard rod either on a level with, or above, the wheel axes, the height of the roll axis is so near to that of the centre of gravity that anti-roll devices are not necessary on a sports car. To illustrate this more clearly a comparison of roll angles for two similar vehicles, one employing single-transverse-link front suspension and the other a double-transverse-link front suspension, both with a through-axle at the rear, is given at the end of this article. The vehicle with the single-link

front suspension is the Allard Palm Beach model, and the calculations are based on data contained in the article entitled "Roll Angles," by Professor Dr.-Ing. Robert Eberan v. Eberhorst, published in *Automobile Engineer* in October, 1951. A value of unity is assumed for the coefficient of friction μ.

The single-link arrangement has only half the number of bearings employed in most of the double-link ones, and only a third of the number in some other layouts. This, and the fact that the radius rod is generally arranged at an angle of more than 45 deg to the link to give about 16-18 in between the centres of the pivot bearings, ensures freedom from serious lost motion, which otherwise might be experienced particularly on braking. Furthermore, it enables rubber bearings of low rate to be employed to give good insulation from road vibrations. Because of the simplicity of the single-link system, and therefore low tooling costs, it is ideal for small quantity production.

Coil springs are most suitable for conventional passenger cars fitted with this type of suspension system. It would not normally be possible to use torsion bars extending from the link pivot centres, because they would have to pass right through the engine sump. To mount them on the frame side members, with an arm and coupling link to each, would considerably increase the number of parts and the weight of the system. A transverse leaf spring is unsatisfactory because, even if there is room to fit a spring which has the desired static deflection and which is not overstressed, its weight would be some 50 per cent greater than that of a pair of moderately stressed coil springs designed for the same static deflection. The effective front suspension frequency obtaining on the current Allard models is between 65 and 70 oscillations per minute, that is, approximately the same as that of the Ford Consul and Zephyr.

The new Ford vertical leg and swinging link suspension system gives, by virtue of its geometry, a high roll centre, but no more than one or two degrees of wheel tilt, Fig. 6. The amount of wheel tilt depends, of course, upon the proportions necessary for the vertical leg to accommodate the travel of the road wheel from the bump to the rebound positions. This system cannot be adopted easily on a vehicle with a separate chassis, but with a stressed body structure, as on the Ford models, it is ideal; for it makes possible a considerable saving in weight and is inherently more rigid than most double transverse wishbone link arrangements.

Roll angle calculations for split axle front and beam axle rear suspension

$$\text{Roll angle } \Psi = \frac{\mu W\left[h - \frac{an+bm}{C}\right] + \left[\mu W_1 (m-r)\right]}{\frac{1}{2}\left(\frac{b_2 t_2}{f}\right)^2 C_1 + 2C_2 a_2^2}$$

Dimensions and loads for the single transverse link front and beam axle rear suspension of the Allard Palm Beach:

W_1 = 130 lb = unsprung mass (front)
W = 1,470 lb = weight of sprung mass
C_1 = 140 lb = front spring rate
t_2 = 49 in = rear track
f = 23·5 in = length of swing axle
b_2 = 15·25 in = spring to pivot centres (front)
a_2 = 17 in = chassis centre-line to spring centre (rear)
h = 20 in = centre of gravity above ground
a = 41 in = centre of gravity to front axle
n = 16·5 in = height of rear roll axis
b = 55 in = centre of gravity to rear axle
m = 10·75 in = height of front roll axis
c = 96 in = wheelbase
r = 12 in = rolling radius
C_2 = 56 lb = rear spring rate
μ = 1 (hypothetical)

$$\Psi = \frac{\mu W\left[h - \frac{an+bm}{c}\right]}{\frac{1}{2}\left(\frac{b_2 t_2}{f}\right)^2 C_1 + 2C_2\, a_2^2 + \left[\mu W_1 (m-r)\right]}$$

$$= \frac{1{,}470\left[20 - \frac{(41\times16{\cdot}5)+(55\times10{\cdot}75)}{96}\right]}{\frac{1}{2}\left(\frac{15{\cdot}25\times51}{23{\cdot}5}\right)^2 140 + (2\times56\times17^2) + \left[130(10{\cdot}75-12)\right]}$$

$$= \frac{1{,}470\times6{\cdot}8}{\frac{1}{2}(153{,}500)+325{,}000-162{\cdot}5}$$

$$= \frac{10{,}000}{401{,}588}$$

$= 0{\cdot}0249$ radian

Therefore $\Psi = 1$ deg 26 min.

Dimensions and loads for the same car, but with double transverse link front and beam axle rear suspension. The roll centre is assumed to be at ground level:

a_1 = 22½ in = roll centre to outer pivot, lower
b_1 = ∞ = instantaneous centre to outer pivot point
d_1 = 25½ in = roll centre to tyre contact point
c_1 = ∞ = instantaneous centre to tyre contact point
C_1 = 52 lb/in = effective front wheel rate
t_1 = 51 in. = front track

Without control bar: $\mu W\left[h - \frac{an}{c}\right] + (\mu W_1 r) = M_1 + M_2$.

For independent suspension with horizontal parallel links:

$$M_1 = \frac{t_1^2}{2} C_1 \Psi$$

$M_2 = 2C_2\, a_2^2\, \Psi$, as before.

Therefore:

$$\Psi = \frac{\mu W\left[h - \frac{an}{c}\right] + \mu W_1 r}{C_1 \frac{t_1^2}{2} + 2C_2\, a_2^2}$$

$$= \frac{1{,}470\left[20 - \frac{41\times16{\cdot}5}{96}\right] + 130\times12}{52\times\frac{51^2}{2} + 2\times56\times17^2}$$

$= 0{\cdot}051$ radian

$\Psi = 2$ deg 54 min.

RITOLASTIC

A NEW pamphlet, entitled "Ritolastic," which is the name of a liquid bituminous coating obtainable with colour finishes, has recently been published by Andrew Maxwell (The Liverpool Borax Co. Ltd.), St. Paul's Square, Liverpool 3. Copies of these pamphlets are available on application to their Technical Service department. Black or aluminium coatings do not always harmonize with surrounding colour schemes, and under these circumstances, pigmented bituminous coatings may be used to advantage. This treatment is said to give protection against the action of salt and other corrosive materials often flung on to the underside of motor vehicles, and is effective in preventing deterioration of timber or metal components.

PNEUMATIC TOOLS

A VAN, carrying samples of the range of pneumatic tools supplied by the Atlas Diesel Co. Ltd., of Wembley, is visiting factories all over Great Britain to give demonstrations. This van is intended primarily for visiting engineering works where pneumatic tools such as grinders, chippers, riveters, drills, scalers, etc., are used. It carries a full range of these tools as well as specialized ones, such as tappers, screwdrivers, nut runners, metal-shears, and portable saws. The aim is at making this equipment available to executives and operatives so that they can see, handle, and sense the balance and feel of the tools. In addition, during the visits, adjustments or simple repairs can be made to equipment already installed in factories.

THE ZF-MEDIA GEARBOX

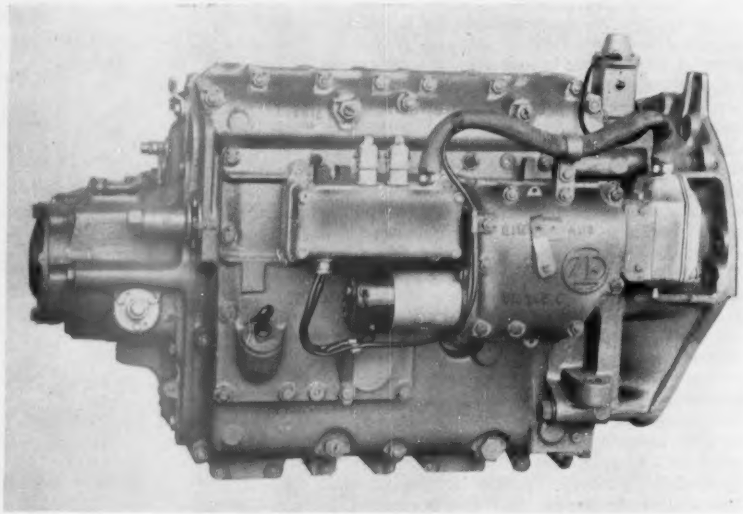
A Six-ratio Unit for High-speed Coaches

FOR many years past the Zahnradfabrik Friedrichshaven A.G. has enjoyed an enviable reputation for the design and manufacture of gearboxes and transmissions of all types. Units for the larger commercial vehicles have always formed an important part of the firm's output and it may be remembered that they introduced, many years ago, a gearbox in which all speeds were engaged by multiple-disc clutches operated directly by electric ring-magnets.

The gearbox to be described is a direct development of the electro-magnetic unit. Experience having shown the importance of safeguarding the vehicle transmission from the consequences of any sudden electrical defect, the makers decided that the engagement of all the clutches in the new design should be purely mechanical as far as the box itself was concerned. Electricity was retained solely to give remote and specially modulated control of the mechanism in the gearbox.

This attitude was largely dictated by the experience of the firm in supplying transmissions for buses and coaches operating over the mountain passes of Switzerland. In such service the vehicle engine is invariably used as a brake, its effect being frequently amplified by exhaust back-pressure braking.

Under the new system any electrical failure leaves the particular gear in use fully engaged and no loss of brake power is involved. Provision is also made whereby, with the vehicle at rest, any desired gear or neutral can be engaged by removing a cover, inserting a tommy bar into a capstan head, and rotating the cam drum which operates the multiple-disc clutches. Since—although the gearbox has frictional engagement of all forward gears—an engine clutch or fluid flywheel is still employed, it is quite feasible to drive home a vehicle suffering from some defect in the relatively complicated electrical control gear.



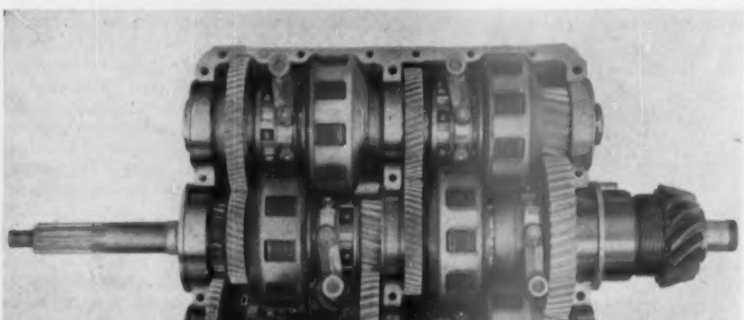
Urban bus box complete with electric control unit

A further advantage of the electro-mechanical system is that the load on the batteries is greatly reduced. In place of the continuous, and considerable, energization current taken by the two magnetic clutches happening to be in use on any given ratio, no current passes during normal running. The fairly heavy current taken by the gear-changing motor lasts for a fraction of a second only.

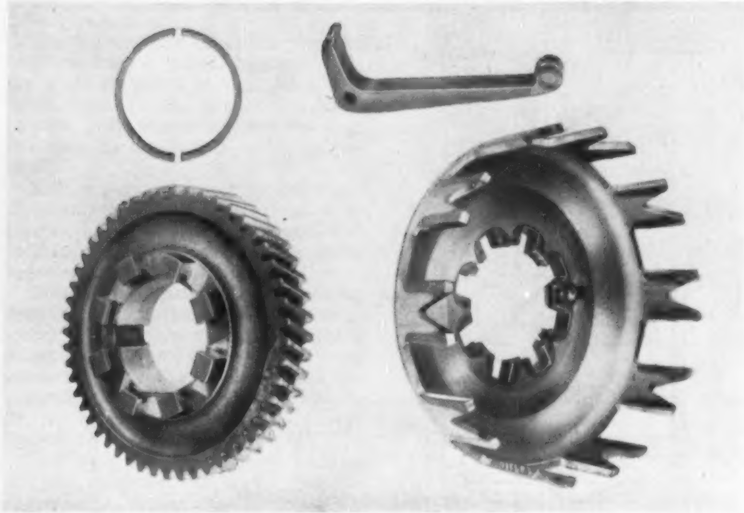
In spite of the fact that a separate engine clutch or fluid flywheel is necessary with the new ZF-Media gearbox, the multiple-disc clutches in the gearbox are fully capable of dissipating the energy associated with an upward or downward change. It is, however, recommended that the start from rest should be made on the engine clutch or fluid flywheel with the appropriate gearbox clutches fully engaged. The mean diameter of each of the five disc clutches is no less than 6·5 in, there are about twelve driving and thirteen driven plates and about ten pounds of steel is available to absorb the heat generated during engagement over a friction surface of about 250 in^2.

While the power flow is momentarily interrupted during a change, the period of disengagement is only a fraction of a second and the kinetic energy of the engine flywheel can be fully utilized in a rapid upward change without necessarily releasing the accelerator pedal during the process; movement of a small lever on the steering column is all that is required—the engine clutch pedal need not be used.

The clutch-operating cam drum in the gearbox determines the changing process, which is sequential and without any possibility of " by-passing " a ratio. In some cases the neutral position is arranged between 2nd and 3rd speeds, while in others (particularly railcars) neutral comes behind 1st speed. Return to neutral when stopping



Six-speed box, with direct top gear, for rear-engined urban buses



Clutch components

the vehicle is preferably done after coming to rest; for obvious reasons a rapid change down through all the gears is highly undesirable while the vehicle is in motion.

On railcars, which are always equipped with a freewheel in the transmission, there is no possibility of damaging the engine or the gearbox by abnormal overrun speeds when changing down. For road vehicles, however, a special centrifugal governor, driven from the input shaft of the gearbox, interrupts the electrical circuit and halts the downward change until the rotational speed of the input shaft (and engine) has fallen to a value that will make the next drop step of 1·54 : 1 quite safe.

On account of the space taken up by the five disc clutches the gearbox is large and heavy. The two layshafts are at about 6·25 in centres, the box is about 2 ft long, and the weight with an aluminium casing is 551 lb. Nevertheless, the ZF gearbox is well suited to the long-distance, high-speed coaches now so widely used on the Continent, since it provides six speeds, evenly graduated over a range of 8·66 : 1, and both upward and downward changes can be made shocklessly without special judgment on the part of the driver.

The wide range of ratios enables a top gear to be provided that will keep engine revolutions low when travelling for long periods at the high speeds permitted by the "autobahnen." Furthermore, the close ratios (steps of 1·54 : 1 throughout) available in the lower gears, together with almost instantaneous upward changing, should make possible an excellent performance on the long, well-engineered and fairly steep mountain passes of Europe. Ability to make, and hold, an upward change on a gradient after a temporary check may make a difference of as much as 25 per cent in the running time of a coach, compared with that of one which, because of, say, 2 : 1 steps in the lower ratios and an awkward gear-change, is condemned to continue for miles uphill on a gear much lower than circumstances require.

The ZF gearbox has also found wide adoption on normal urban bus services: several hundreds are now running and some of them have covered over 70,000 miles with satisfaction. It is, however, clear that for much of such work the six-speed box is an unnecessary refinement and that the overall range of 8·66 : 1 is not required.

It is understood that the firm are developing a transmission solely for urban buses, in which a torque converter drives through a three-speed box with multiple-disc clutches operated hydraulically and under automatic control. An interesting feature of this design is that the torque converter operates only on the lowest ratio, the power on the two higher gears passing direct from the engine flywheel to the gearbox, so that all converter losses are eliminated during the greater part of the running.

The converter starts the vehicle from rest and brings it up to a speed corresponding approximately with 2nd gear of a four-speed bus gearbox, being responsible for that portion of the acceleration which normally causes unpleasant jerking and places a heavy duty on the engine clutch facings.

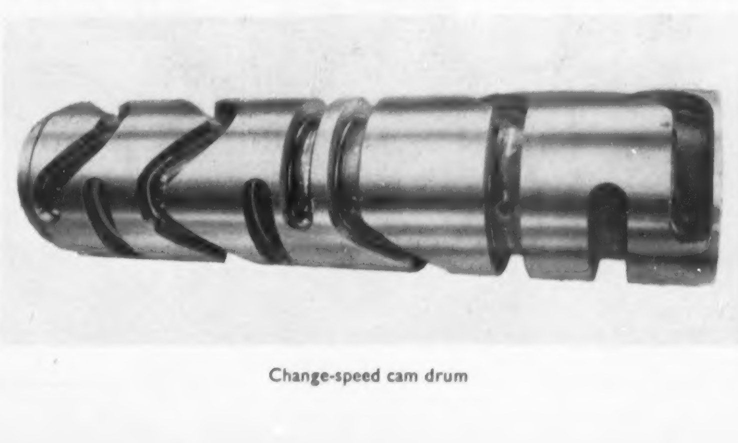
Before passing to a detailed description, it should be mentioned that the six-speed ZF gearbox is available with either direct drive or overdrive on top speed, the only alteration necessary being replacement of the first pair of gears in the box by a similar pair interchanged in position on drive shaft and upper layshaft and a new cam drum. In all cases, reverse is provided by a double dog clutch disengaging one wheel in the 1st-speed train and engaging another wheel driven through an idler. This change is effected mechanically by a hand lever after the vehicle has been brought to rest, final engagement being by moving the electrical control lever to 1st-speed position.

Mechanical interlocking is provided whereby the reversing lever, should the driver have neglected to return it after reversing, is automatically returned to neutral by the cam drum as soon as 2nd speed is engaged. This removes the dangerous possibility of the driver subsequently changing down (electrically) to 1st speed and discovering he has reverse gear instead.

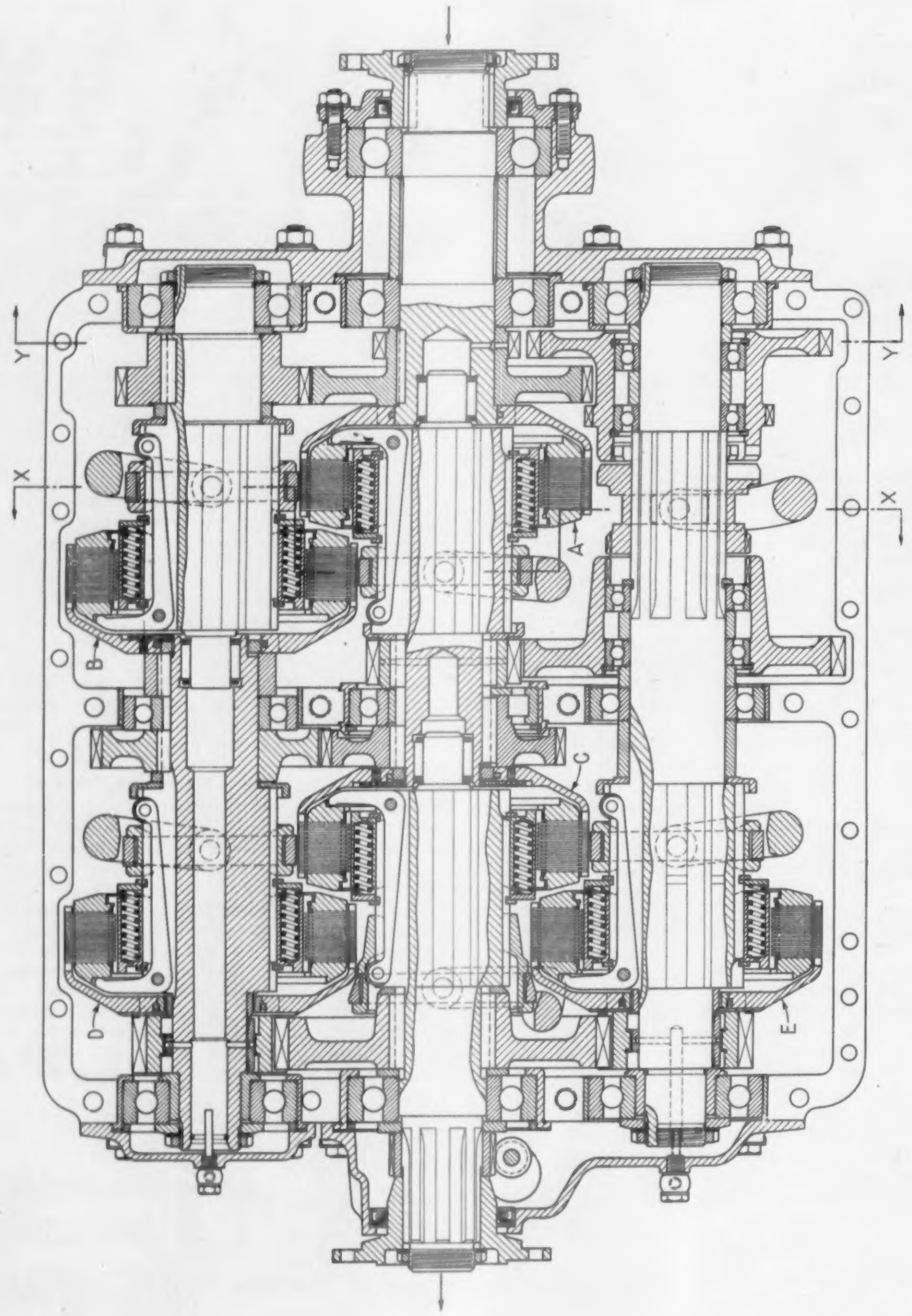
Maximum input rating for the direct-drive box is 50 metre-kilogram, or 362 ft-lb. The overdrive box can accept an input of 75 m-kg, or 543 ft-lb. All steps are in the ratio of 1·54 : 1, so that the direct-drive box has a bottom gear of 8·66 : 1 and the overdrive version has a 6th speed of 0·66 : 1 and a 1st speed of 5·52 : 1. Diesel engine speeds are commonly rather higher in Germany than in Britain and the permitted maximum for each box is 2,250 r.p.m.

Gears and clutches

An overdrive unit is shown in section. The input shaft drives the front half of the upper layshaft by hardened and ground helical gears giving a speed increase of 1·54 : 1. Multiple-disc clutch B transmits the power to the rear half of the upper layshaft when required. On the main axis there are three shafts, the input shaft, the intermediate shaft, and the output shaft. A helical gear keyed to the rear half of the upper layshaft drives, at 1 : 1, a gear keyed to the intermediate shaft which also carries a



Change-speed cam drum



GENERAL ARRANGEMENT OF ZF-MEDIA GEARBOX

Overdrive model for high-speed coaches. Both the earlier ball bearing and the current roller bearing for the central intermediate shaft are shown

pinion driving the lower layshaft at 2·37 : 1 reduction.

The forward end of the intermediate shaft carries the hub of disc clutch A, the shell of which is attached to the input shaft. To the rear of the intermediate shaft is also attached the shell of clutch C; the hub is on the output shaft.

Both upper and lower layshafts can be connected by clutches D and E to pinions idling on roller bearings and meshing, at 2·37 : 1, with a large wheel keyed to the output shaft.

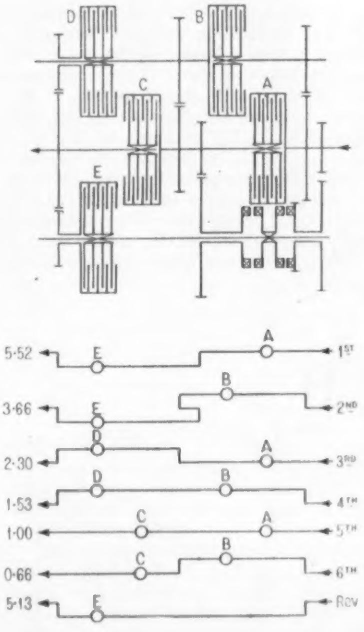
Reverse is obtained by a sliding double dog that disengages the central wheel on the lower layshaft (which forms part of the 1st-speed train) and engages in its stead a gearwheel at the extreme front of the box, which is driven through an idler by a pinion carried by the input shaft.

The power flow is given in the diagram, together with the clutch-engagement sequence. It will be observed that, in order to reduce the maximum torque taken by the clutches, those responsible for the final transmission of the four lower speeds are mounted on the faster-running parts of the train. This means, for example, that on overdrive with a moderate engine speed of, say, 1,800 r.p.m., the shells of clutches D and E are rotating at no less than 1,800 × 1·54 × 2·37 r.p.m., that is 6,570 r.p.m. The hub of clutch D is running at 1,800 × 1·54, or 2,772 r.p.m., while the hub of clutch E is turning at only 2,772 ÷ 2·37 = 1,165 r.p.m.

The relative speed of the two sets of clutch plates in clutch E is therefore no less than 6,570 – 1,165 or 4,405 r.p.m. In prolonged usage of overdrive this might lead to appreciable generation of heat due to the slight, but inevitable, drag of the idle clutch plates. To obviate this the firm arrange, in gearboxes intended for vehicles operating on the autobahnen, that the dog clutch used for reverse is left in the neutral position, so that the lower layshaft is disconnected from its central driving wheel. Thus the hub of clutch E is allowed to pick up speed until it runs at the same speed as the shell and heat generation ceases. This means that only the four higher speeds are available: on entering hilly country the dog clutch would, of course, be engaged at the next stop, making all six speeds usable.

To reduce as far as possible the drag of the idling clutches, half the discs are pressed in a slightly wavy form after hardening, grinding and lapping. The waves are only about 0·1 mm deep but are very effective in breaking up the surface contact on release of the clutch.

The inner plates of the clutch (there are twelve or thirteen in number) slide on splines on a hub carrying an abutment plate located by a shoulder on the splines. The hub assembly slides on splines on the shaft, being held against a circlip on the shaft by a number of preloaded compression springs housed in drillings in the hub. The clutch presser plate is also splined



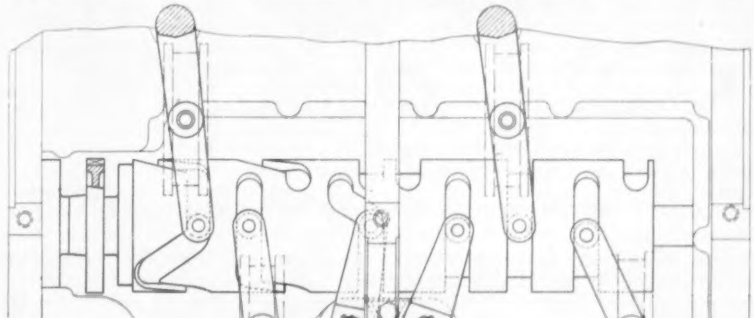
Diagrammatic arrangement of overdrive box and clutch engagement sequence for all ratios

to the hub and is operated by five bell-crank levers, whose horizontal ends are provided with rollers which are thrust inwards by the bevelled edge of an overriding clutch-operating muff.

In the fully engaged position the rollers are inside the parallel bore of the muff and therefore exert no end pressure on it or on the striking collar. Engagement of the clutch is cushioned, first by the waves on the discs, secondly by the elasticity of the bell-cranks, which are made as slender as possible, and finally by the yielding of the pre-loaded springs in the hub. These springs also perform a most valuable function in taking up any slight wear of the plates. No adjustment mechanism, therefore, is necessary. The total travel for disengagement in the stack of discs is only about 3 mm, and it has been found that increased clearance leads to the discs running out of true and causing greater wear than when kept fairly snug.

Clutch hubs are hardened by heat-treatment but the clutch shells, which are machined from heavy pressings and gashed to accommodate the lugs on the discs, are flame-hardened on the edges of the slots. Each shell is driven from its adjacent wheel by milled dogs on the latter entering broached notches in the shell. An internal groove is turned in both parts and a divided ring, kept in place by a solid ring inside it, gives axial connection of the two parts. Flame-hardening is given to the edges of the broached notches also.

The general construction of the gearbox follows accepted practice in heavy-vehicle units. All three shafts have intermediate bearings, these taking the form of single-row ball bearings housed in steel sleeves fitted to the split aluminium-alloy casing. The latest boxes have a parallel-roller bearing for the central intermediate shaft, this being provided with shoulders in both races to take the end-thrust set up by the helical gears. The idling bearings of the two pinions at the rear of the layshafts also have two rows of parallel rollers, each row taking

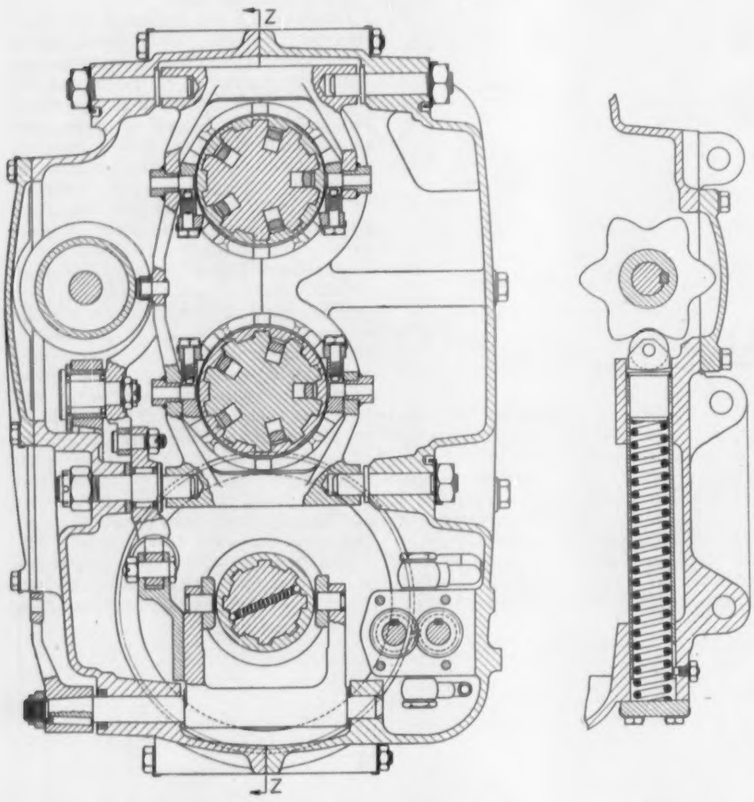


Cam drum and striking gear. (View on Z-Z)

end thrust in one direction. These bearings are, of course, only subject to end thrust when the adjacent clutch is engaged and the whole assembly is rotating as a mass without relative movement between rollers and races.

The intermediate shaft bearing, however, has to take appreciable end thrust when running at high speed. In Britain the successful employment of parallel-roller bearings taking end thrust in hubs and the like is familiar but some suspicion might be aroused by this arrangement applied to a high-speed journal. It should, therefore, be mentioned that Continental manufacturers have for years been successfully making helical constant-mesh gearboxes in which each wheel is individually carried in the casing by pairs of single-row roller journals, one of which takes the end thrust. There are three spigot bearings in the box and all are identical in size. They have long parallel rollers, caged and bearing on races ground directly on the components.

As previously mentioned, reverse is had by a train of straight spur gears with the usual idler, the driven wheel idling on ball bearings on the lower layshaft and being clutched to it at will by a manually operated dog member. On railcars, in which reverse is had by alternative bevel gears, the reverse gear in the box is not required and the dog is omitted. The reverse gear train is, however, still fitted, as it serves to drive a lubrication pump. This gear-wheel pump delivers oil by open jets axially into the bores of the rear ends of the two layshafts, from which the oil is flung radially through drillings, to the idling roller bearings of the two pinions which, as already mentioned, revolve at exceptionally high relative speeds on overdrive and, therefore, call for special treatment. The boring in the upper layshaft is continued to supply oil to the spigot roller bearing where the shaft is divided. Neither of the two spigot bearings on the centre shafts have a pumped supply, the oil sprayed inwards from radial holes at the bottoms of the meeting teeth of the adjacent gearwheels being found adequate.



Clutch-operating mechanism. (Section on X-X and scrap section on Y-Y)

The box described has the layshafts arranged in a vertical plane but, where circumstances make it more convenient, it can be mounted with the shafts in a horizontal plane. The oil pump, which comes in one corner of the casing, does not require re-positioning, as it is below the oil level in both cases. Standing oil level is to within about ½ in of the centre line of the lower layshaft and sufficient agitation is set up to lubricate all the gears with spray. Extreme-pressure oils are advised and, in view of the drag of the disc clutches, a lower viscosity than is commonly used in gearboxes is necessary. Summer oil should be SAE 30 and winter oil SAE 20, or even SAE 10 in very cold climates.

Clutch-operating gear

In the latest version of the ZF gearbox the muffs of all five disc clutches are slid into position by yokes, split and secured by studs and completely enveloping the muffs. The yokes are carried by two pins each in the arms of the swinging striking forks and are liberally drilled with large holes to allow free oil spray access; they are necessarily bored full large to permit of the arcuate motion of the striking forks.

The dog clutch for reverse, and for cutting out 1st and 2nd speeds to eliminate drag losses in clutch E, is operated by a swinging striking fork carrying pivoted rectangular slipper blocks engaged in the groove of the sliding member. This striking fork is connected by an external lever and rodwork to a hand lever in the vehicle cab. A special groove in the change-speed cam drum is also connected by somewhat intricate mechanism, including a preloaded spring, to the reverse striking fork. It operates positively to disengage reverse when the cam drum is moved to 2nd-speed position, taking the hand lever with it so that when next changing down into 1st speed (which is used also as part of the reverse train) the reverse gear will not be engaged unless the driver deliberately moves the hand lever again into the reverse position.

The muffs of all five disc clutches are operated by rollers, turning on needle bearings, working in cam grooves end-milled in the hollow steel cam drum which turns on two ball bearings. One end of the cam drum shaft projects from the rear of the box and carries an indicator disc and a hexagon end to which a spanner can be applied to turn the drum. This is of value in putting the box "through its paces" on the test bed before the electrical operating gear has been fitted; it can also be used on the road to engage any desired speed in case of failure of the electrical mechanism.

In a normal road vehicle gearbox the cam drum has seven equally spaced stations corresponding to the six speeds and a neutral position, when all disc clutches are disengaged. A roller-ended locking plunger with a very strong spring works in seven deep notches in a disc secured to the cam drum shaft.

In railcar applications where a fluid flywheel and a free-wheel, together with a separate reversing box giving its own neutral, are used, there is no neutral position on the cam drum, which has six positions only and has a six-notched locking disc. This has curved notches with sharp peaks between and performs a useful function in reducing the maximum load on the gear-changing motor. It will be understood that the first half of the movement from one speed to another produces merely the withdrawal of one or two clutches, an operation which

requires no appreciable effort at the cam drum and may even supply energy as the roller ends of the bell-cranks fly outwards over the bevelled edges of the muffs. The small series-wound motor can, therefore, be safely called upon to compress the heavy plunger spring and store up energy which is released to assist the motor as the plunger roller rides over the tip of the peak during the second half of the movement, when the engagement of one or more clutches makes a heavy demand on the operating mechanism.

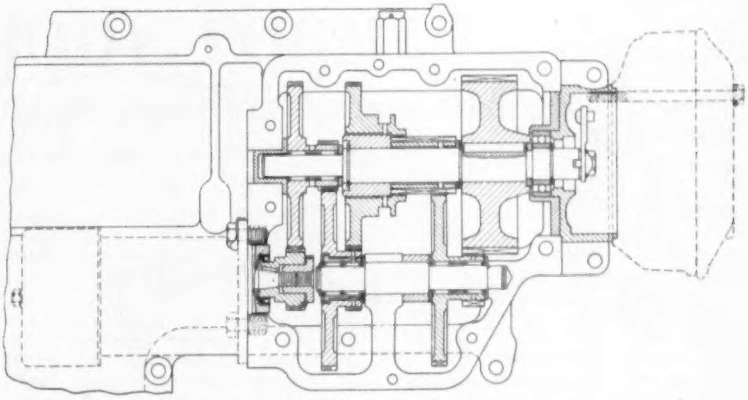
Normal operation of the cam drum is by the electric motor driving through a five-stage spur reduction gear giving an overall reduction of 318 : 1. Provision is made, by an external level on the box sliding one of the gears out of mesh, for disconnecting the motor in the event of electrical failure.

Included in the train is a coupling with a bolt working in a clearance hole, which gives several turns backlash. Thus the plunger spring, when nearing the final position, is able to drive the cam drum ahead of the motor, allowing it several turns to come to rest under electrical braking, after the cam drum has stopped in any gear position. This removes the possibility of the kinetic energy of the motor armature taking the cam beyond the correct position and setting up an oscillatory movement.

Final drive to the cam drum from the unit comprising the electric motor and reduction train is by a large spur gear housed in the reduction gear casing and meshing with a similar one on the cam drum. The spindle of the first-mentioned gear carries an arm working the contact arms of the electrical switchgear. The whole of the electrical mechanism and reduction gear forms an independent unit which can be withdrawn sideways from a facing on the gearbox.

Electrical control gear

Although the mechanism to be described gives neither pre-selection nor automatic gear-changing, its considerable complexity is more or less necessary for two reasons. In the first place it provides remote control without mechanical connections, except for reverse; underfloor engines, rear engines, or railcars with double-ended control can be dealt with by using a suitable length of 10-conductor cable.



Cam-drum drive reduction gearing

Secondly, power operation of some type is in any case essential, since the effort of engaging the disc clutches in the gearbox is greater than a driver can conveniently exert.

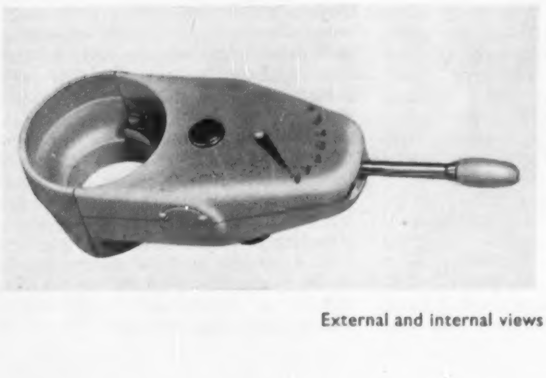
Carried by a split casing clipped to the steering column is a change-speed lever; it is normally held in a mid-position, extending laterally, by double centering springs. When moved forward to the limit of its travel a pawl attached to it moves a contact finger from one stud of a set of seven to the next. Assuming the contact finger to be initially on the zero or neutral stud, full forward travel of the lever will put the contact finger on the stud corresponding to 1st speed, energizing the lead controlling this speed in the mechanism on the gearbox.

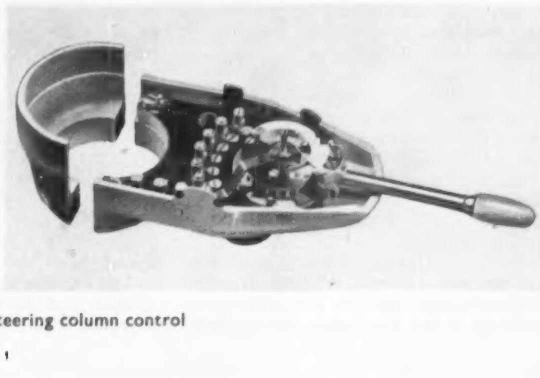
The engagement of 1st speed is effected by the electric motor, the motion of which is rapid at first to take up clearance but is automatically slowed down to a reduced speed to give a smooth final engagement. The driver can, of course, cushion the actual start by using the pedal-operated engine clutch, which is normally fitted to road vehicles; most railcar applications include a fluid flywheel. This is the normal procedure when starting from rest, but when making subsequent upward changes the gearbox clutches are usually allowed to perform the whole duty.

Only one upward step can be made at a time and all gears must be gone through to arrive at top speed. There is, however, nothing to prevent the driver from "ratcheting up" several gears in quick succession—the motor merely follows the "instructions" as fast as it can. For example, while the vehicle is at rest with the engine clutch disengaged, the driver can make three forward movements of the lever in succession and then start on 3rd speed by letting in the engine clutch. A pointer on the spindle carrying the contact finger shows its position, while an indicator light shows when the electric motor mechanism in the gearbox has actually completed its "instructions."

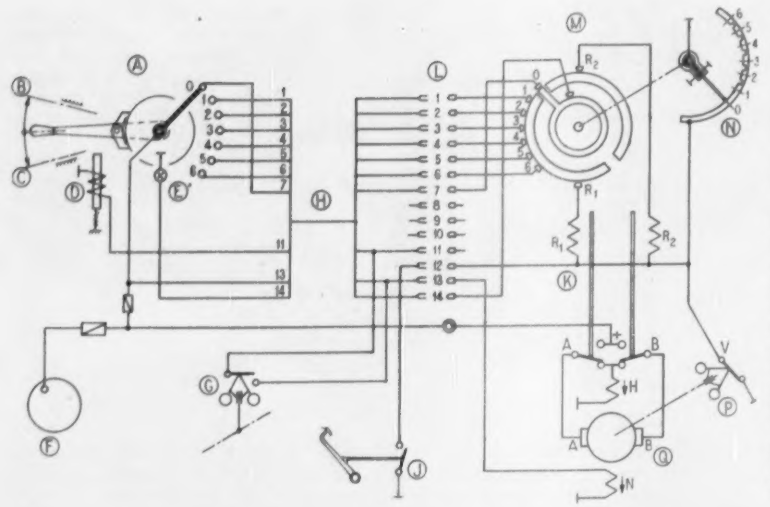
For downward changes the operation is similar but, for road vehicles not equipped with a freewheel in the transmission, the firm recommend and can supply a centrifugal governor, driven either by the engine or the input shaft of the gearbox. This brings into action a pawl inhibiting movement of the contact finger in a downward direction until engine speed has fallen to about 75 per cent of maximum, thus preventing dangerous overspeeding of the engine by a too rapid change down.

On the other hand, when the main clutch has been withdrawn and the engine is idling, a switch operated by the pedal cuts out the special modulating control which normally slows down the final engagement of the gearbox clutches. The speed of downward change is thus actually increased beyond normal, to enable the gear to be brought back to neutral as quickly as possible when making a stop. The contact finger on the steering column is connected to battery positive, and each of the contact studs numbered 0





External and internal views of steering column control



A, Change-speed controller: B, Up change: C, Down change: D, Locking-pawl solenoid: E, Indicator lamp: F, Battery positive connection: G, Over-run governor: H, 10-lead cable: J, Clutch-pedal switch: K, Double relay: L, 14-point connector: M, Segmental contactor: N, Modulator contactor: P, Modulator governor: Q, Change-speed motor

Electrical control. Diagram of connections

to 6 on the diagram, with the 0 point called 7 in the listing of the connecting leads, is connected to similarly arranged spring contacts which bear on two segments, separated by a narrow lug connected by a slip ring and lead 14 to the indicator lamp on the steering column. The segments are coupled to the cam drum in the gearbox and take up corresponding positions. Each segment is connected respectively by spring contacts R1 and R2 to the corresponding windings of a double relay.

Assume, for example, that the contact finger on the steering column is at 0 and that the lug between the segments is in contact with spring contact 0. Current will pass through the contact finger and the lug to the indicator lamp, which will show the driver that the gearbox cam-drum is in the neutral position. If the driver now ratchets the contact finger to position 1, current to 0 will be interrupted and will pass instead, by way of spring contact 1, to the lower segment and thence to winding R1 of the left-hand relay. This connects moving contact A to battery positive, contact B being earthed through the series field winding H of the change-speed motor. Current then passes through the armature and series field, which latter is supplemented by a subsidiary field N, taking about 12 watts and permanently in circuit.

The motor then starts and revolves the cam drum and with it the segments until the lower one breaks contact with spring contact 1, which is left in contact with the lug supplying the indicator lamp. The current to R1 being interrupted, the left-hand relay contact drops, cutting off the current supply and short-circuiting the motor. Its motion is quickly arrested as the subsidiary field N is always energized.

Movement of the contact finger on the steering column back to 0 would energize R2, since the spring contact 0 is then on the upper segment, the right-hand relay would be energized, reversing the motor and bringing the cam drum back to neutral.

So far the unmodulated operation of the gearbox has been described, with the electric motor running at full speed during the change. This would result in a rapid release of any gear that was engaged and an equally rapid engagement of the next one; the first is desirable to save time but rapid engagement would cause excessive shock. Instead of R1 and R2 being directly earthed, they are in fact earthed through a sliding contact carried, but with some backlash, by the segment assembly, and sliding over a slip-ring with notches cut in it in between each gear position.

When changing, say, from 1st to 2nd gear, the sliding contact lags behind somewhat and drops into the notch well after the midway position of the cam-drum. That is, when all clutches are fully released and the engagement of the next one has commenced. When the sliding contact drops into the notch the earth connection of R1 is broken and the relay cuts off the motor current.

When the motor has slowed down considerably, a second earth becomes available through contact V, which is controlled by a small centrifugal governor P on the electric motor spindle, adjusted to make contact when the speed has dropped considerably.

The alternative earth thus provided re-energises the relay R1 and the motor carries on to complete the engagement of the gearbox clutches at a speed which will give a smooth pick-up.

Yet another alternative earth to the relay windings is provided by the clutch-pedal switch J. By short-circuiting the modulating governor and sliding contact, this switch allows the electric motor to run at full speed throughout a change, thus quickening the operation of bringing the gearbox to neutral when stopping the vehicle.

ALTERNATOR SYSTEMS

IN a paper entitled "Why Alternator Systems?" by A. D. Gilchrist, *S.A.E. Preprint*, August 17-19, 1953, it is stated that when selecting a generator size for a particular vehicle, the summation of the total electrical load in amperes is found to be unreliable. Therefore, it is recommended that the time factor be included by comparing the ampere-hour loading for all accessories with the ampere-hours generated within a given time cycle. An ampere-hour loading of as much as 80 per cent of the ampere-hours generated may safely be used.

A comparison is made between a 14 volt, 50 amp, D.C. truck generator system, and a 14 volt, 60 amp, rectified A.C. alternator system. The three main sections of the generator, that is, field, armature, and commutator and brushes, may be compared respectively with the rotor, stator and rectifier in the alternator system. Commutation is the main limitation in obtaining a wide speed range in the generator, since low-speed performance requires a large number of turns per armature coil, with resultant limited brush life. The maximum speed of the D.C. generator is thus about 5,000 r.p.m.; the alternator, having a much lower rotor current fed by slip rings, can be safely run up to 12,000 r.p.m. With suitable pulley ratios, this may give a good full-load performance curve right down to idling speed.

The rectifier used with the A.C-D.C. system is of the three-phase, full-wave, dry-plate selenium type. In the D.C. system, a three-element voltage regulator, having a reverse current cut-out, is employed. This regulator also incorporates a load or current limiter, to protect the generator from overloads during low-gravity battery operation, and an ordinary vibrating-contact voltage regulator. The unit for the A.C-D.C system has also three functional elements, but the first is a load relay, which differs from a cut-out in having no series winding. It is energized through the ignition circuit and closes or opens the main D.C. load circuit from the rectifier. The A.C-D.C. system costs about 50 per cent more than the D.C. system at present, but the weight of the machine and control unit is only 34½ lb as compared with 47½ lb. *M.I.R.A. Abstract No.* 6507.

BODY FORM

The Aerodynamic Aspects of Modern Styling

J. P. Milford Reid, A.F.R.Ae.S., M.S.A.E.

IN Britain, although the importance of aerodynamics is now more widely appreciated, improvements made are usually incidental to revised layout and cleaner styling. By contrast, most Continental manufacturers initiate new projects with wind-tunnel tests on models. Their value is illustrated by the Dyna 54 (drag coefficient $C_x = 0{\cdot}21$) which seats six on two 53 in seats and attains 81 m.p.h. fully laden on 40 b.h.p. from an 850 cm³ twin-cylinder engine. At 50 m.p.h. it achieves 40 m.p.g. using only 16 h.p. Acceleration to 50 m.p.h. in ten seconds results partly from light weight (11 cwt) but low drag contributes much to reaching 75 m.p.h. in 24 sec. Expense prohibits aluminium construction in this country, but a steel four-seater of comparable shape could have similar acceleration.

Aerodynamic efficiency and appearance are both improved on the Porsche by the use of inclined headlamp glasses

Grouping the entire transmission at the front permits a flat floor, so improving the drag coefficient besides eliminating chassis depth from the frontal area. This is equally practical with rear engines, and the Porsche 356 achieves 87 m.p.h. on 40 b.h.p. from a 1·1-litre engine and 106 m.p.h. on 70 b.h.p. with the 1·5-litre unit. On the Renault Frégate ($C_x = 0{\cdot}25$) independent rear suspension and a divided propeller shaft reduce height and frontal area, incidentally permitting the retention of a spiral bevel axle having a low friction loss as compared with that of a hypoid axle.

By sloping the headlamp glasses, as on the Volkswagen and the Porsche, negligible drag is induced, whereas the projecting vertical lamps on the Triumph Sports consume several horsepower, because the airspeed over the well-rounded nose travels 50 per cent faster than the road speed. Mass-production of an inclined lens is said to be difficult, but with the 2 ft 6in height regulation, the possibility of obtaining an additional 3 m.p.g. is worth intensive development.

The standard headlamp location in the wings also reduces the drag saving of a falling bonnet profile by obstructing air spill over the bonnet sides. The Jensen 541 illustrates this, the headlamps projecting well beyond an efficient bonnet and good intake location, and the air is thus directed against the screen corners which must have a large radius in consequence. This is more important than rake; the Simca Aronde was projected with a 60 deg screen, but tunnel tests proved that 40 deg was better, with radiused corners. Compared with a design having separate wings, bonnet and headlamps, such as the M.G. 1·25-litre, the lower drag of the Aronde is revealed in average fuel consumption. At steady speeds from 30 to 60 m.p.h. the rates are exactly equal, but the overall averages are 29 and 34 m.p.g. because of lower power requirements for accelerating and hill climbing.

The better front end of the Magnette, with the elimination of wing gullies on the full-width body, reduces drag considerably, in spite of the retention of an upright radiator shell. Careful catwalk design enables this to be retained on a low-drag design such as the Bentley Continental ($C_x = 0{\cdot}37$), on which the headlamp installation is also notable.

The low, horizontal air intake is more efficient, but nearly all designs disguise a vertical film block which determines the bonnet contour and needs a deflector plate to direct the air up on to the matrix. The Standard 8 shows this, while the Triumph Sports reduces height by locating the filler on the engine instead of on the header tank.

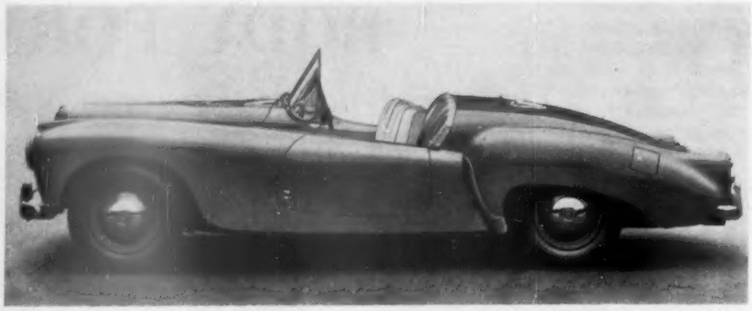
On the Frazer-Nash Hardtop, which has a good air intake, the front end incurs high drag as a result of the exposed wheels and also the headlamp mounting. Ducted cooling systems with low film blocks and separate header tanks are a probable development. The cooling system commonly creates much drag, which ducting could reduce or even eliminate.

Body sides are unimportant. The broken wing lines of the Bentley Continental, Jaguar Mk. VII and Magnette create, perhaps, 2·5 per cent more drag than the slab sides of the Riley Pathfinder or Singer 1500, the mere mouldings of the A.30 and Standard 8 virtually none. The B.M.W. does 28 m.p.g. at 75 m.p.h. in spite of hollow sides, and 24 m.p.g. at 90 m.p.h. with

The Standard Eight gains from the absence of interference drag between separate bonnet, lamps and wings

six passengers, due to a good nose shape.

Similarly, the rear is not critical. The broken roof lines of the Dyna, Frégate and B.M.W. will be noted, but they are well curved in plan. Alternatively, a sharp cut-off may be used, as on the Borgward Hansa 2400, where an increase in base drag is cancelled by reducing the vortices of a tapering tail. This model ($C_x = 0{\cdot}24$) attains 93 m.p.h. with a "normal" consumption of 29 m.p.g. German normal consumption is measured at two-thirds maximum speed, fully laden, with 10 per cent added.



Small fins on the rear wings of the Daimler Conquest Roadster contribute to directional stability

Direct comparison between full-width bodies and their traditional predecessors is difficult because other changes are made at the same time. On the Mercedes-Benz 180 the same 1·25-litre engine developing 52 b.h.p. as in the four-seat 170 S is fitted. Despite increased rolling resistance due to 13 in e.l.p. tyres and 1·5 per cent greater area, normal consumption—with six passengers—is lowered 14 per cent to 32·5 m.p.g. and maximum speed increased 5 per cent to 78·5 m.p.h., while an 8 per cent gradient can be climbed on a top ratio of 3·89 : 1. Most Continental designs are fitted with an overdrive top ratio, and full-width designs are now being fitted with overdrive units in this country, to take advantage of the drag reduction, which varies between 20 and 40 per cent. Possible fuel savings are often disguised by softer tyres, increased area and weight and the higher performance obtained on the road.

TYPICAL C_x VALUES

Model	C_x	Comment
Flat plate	1·25	For comparison
Triumph 1800 Roadster	0·84	Interference between separate parts
Mayflower and Renown	0·68	Razor edges and flat panels
Vanguard	0·65	Bluff nose and rear
Triumph Sports	0·61	Open cars higher by 25 to 40 per cent. Projecting headlamps
Triumph Sports, racing trim	0·55	Cockpit and rear wheel enclosure, no screen
Bentley Mk. VI Saloon	0·54	Large size, good l/d ratio
Humber Hawk	0·47	Full-width design
Nash Ambassador	0·43	Envelope form, curved screen, enclosed wheels, length
Bentley Continental	0·37	Good front, buried headlamps, curved screen, length. Developed in wind tunnel
Renault Frégate	0·25	Developed in wind tunnel
Borgward Hansa 2400	0·24	Developed in wind tunnel
Panhard Dyna 54	0·21	Developed in wind tunnel
Ultimate practical form	0·12	"Modified teardrop"

$$\text{Drag} = \frac{C_x.A.V^2}{400}\,\text{lb}$$

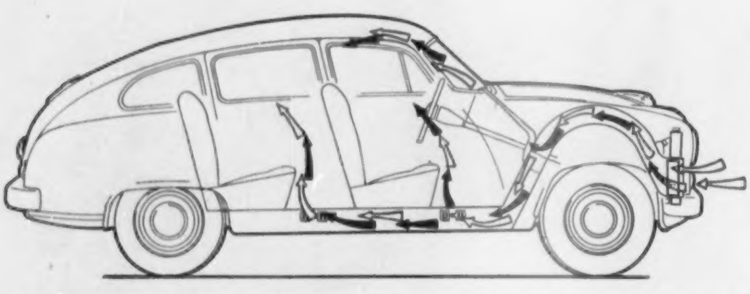
$$\text{Drag h.p.} = \frac{C_x.A.V^3}{150{,}000}$$

where C_x = drag coefficient (dimensionless)
A = front area—ft^2
V = airspeed—m.p.h.
Thus $C_x = 400.k$
where k = form constant used in car performance calculations
C_x is used in preference to k since comparisons are easier.

Full-width bodies are more susceptible to side winds, however, and there is increasing interest in fins. Various designs are seen on nearly all American designs and on the Bentley Continental, Daimler Conquest Roadster, and Bristol models. On the Pegaso Thrill Berlinette they extend upwards into the roof at the rear quarters, the object being to maintain laminar flow around the lee quarter in a cross wind. If they do so, performance will also benefit, since a car invariably travels "out of wind" on the road. They were developed by Touring, the coachbuilders, in the wind-tunnel. Italian body builders frequently test high performance designs for function as well as beauty.

Ventilation and wind noise problems are increasing with the closer airflow past the windows with better front ends and curved windshields. Open windows cause draughts, and on some well-sealed bodies, an organ pipe resonance of about 10 c.p.s. occurs which is most unpleasant. Hinged front ventilators cause whistling and create a depression which attracts dirt and fumes. The ultimate solution appears to be fixed windows and ventilation by means of a ramming front intake with thermostatically controlled heating.

Aerodynamic lift is another problem of growing importance. Forces of over 300 lb have been recorded at top speed on several fast cars. The effect on steering differs with the geometry. On the Allard models, the camber change is clearly visible at speed, while on the Daimler Conquest the steering becomes lighter because castor is reduced. Normal design evolution has thus produced numerous aerodynamic problems needing better understanding and rapid solution if progress is not to be retarded in the production of really low-drag automobiles with the immense advantages in performance and economy which they confer.



On the Borgward Hansa 2400 a vertical rear end increases base drag but, by reducing lift, the induced drag is lowered by an equal amount. To avoid a depression internally, the front air intake is used for normal ventilation

DODGE 7 TON TRUCK CHASSIS

A Vehicle of British Design Powered by the Perkins R6 Diesel Engine

THE design of the new, Dodge 7 ton truck chassis has been effected entirely at the factory at Kew, Surrey, and of all the major components, only the gearbox bears much resemblance to its American counterpart. With a drop side, timber body, the kerb weight of the vehicle is only 3 tons $14\frac{1}{2}$ cwt. Therefore, it can almost be termed a light 7 tonner; for, unlike most other diesel engined chassis designed for this load capacity, it is not intended for operation at the gross maximum legal limit, which is 12 tons for four-wheelers, but at a gross vehicle weight of 25,000 lb. The dry weight of the chassis and cab alone is 6,542 lb, while that of the engine and flywheel is 914 lb, and the gearbox and clutch weigh 229 lb. In the untrimmed, white condition, the cab weighs 658 lb.

When the vehicle was first introduced at the last Commercial Motor Show, it was powered by a petrol engine; however, the Perkins R6 unit is now fitted. This 5·56 litre, diesel engine was fully described in the October 1953 issue of *Automobile Engineer*. It develops 108 b.h.p. at 2,700 r.p.m., and the maximum torque is 240 lb-ft at 1,600 r.p.m. In the Dodge truck installation, an exhauster is driven in tandem with the injection pump.

An engine installation angle of 2 deg 54 min from the horizontal has been adopted, and considerable care has been taken to isolate the chassis frame from engine vibrations. The rear mounting for the power unit is similar to that employed in the Perkins P6 installation in some of the earlier Dodge designs, and it was illustrated on page 89 of the March 1953 issue of *Automobile Engineer*. On each side of the clutch housing is bolted the centre plate of a rubber-to-metal bonded, double sandwich arrangement. The outer plates, of course, are attached to the frame. This sandwich unit is vertically positioned in a transverse plane so that the engine is relatively free to move under the influence of the torsional vibrations, which are reacted by the rubber in shear. Fore and aft motion is more positively resisted by the rubber in compression. Limit-stops are incorporated at the top of each sandwich unit to restrict the amplitude of torsional vibration.

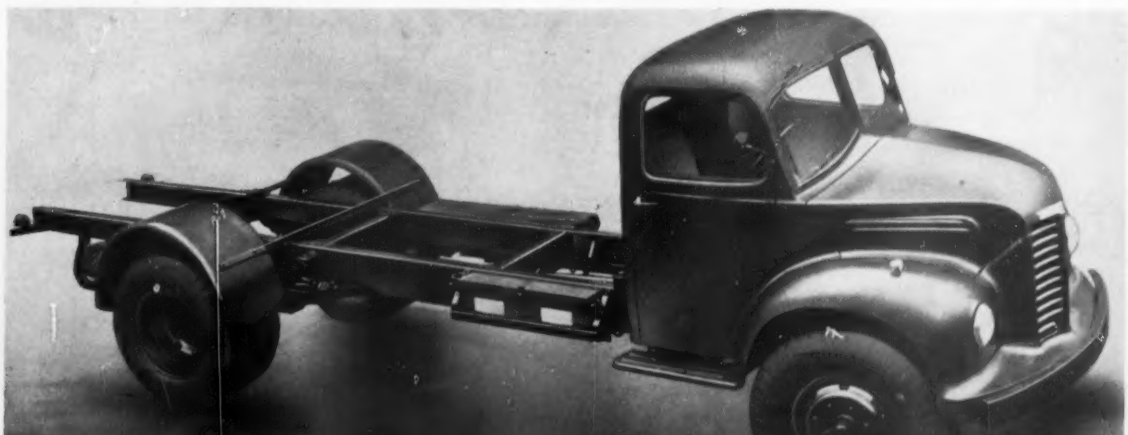
At the front, two rubber-to-metal bonded, double sandwich units are again employed, but the conventional V-arrangement has been adopted. These units are low down on each side of the engine, and they are positioned in such a way that lines drawn, from their centres, perpendicular to their top plates would intersect approximately on the axis of oscillation of the power unit. Heavy steel plates bolted to the frame front cross member are shaped so as to extend over each sandwich unit to act as rebound stops. The advantage of using double rubber sandwich type mountings instead of single ones is, of course, that a relatively large thickness of rubber can be used to give adequate flexibility in shear without there being any danger of instability or excessive bulging of the rubber, due to compression, between the upper and lower plates.

SPECIFICATION

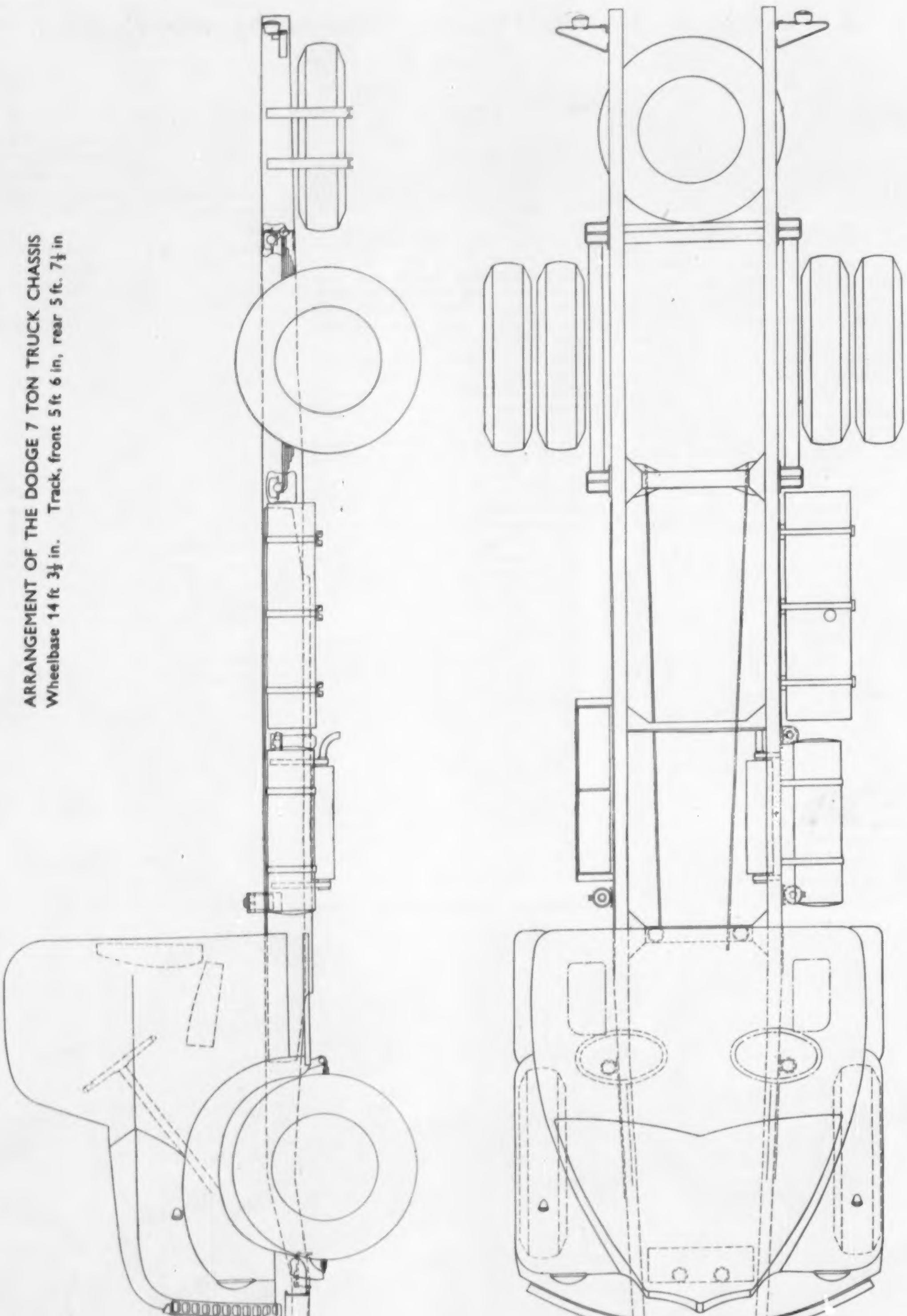
TRANSMISSION: Borg and Beck, single dry plate clutch, type 13AS, 13 in diameter. Gearbox, five forward speeds and one reverse. Ratios, top 1:1, fourth 1·478:1, third 2·395:1, second 4·38:1, first 7·58:1 and reverse 7·51:1. Propeller shaft, Hardy Spicer type 1510.
REAR AXLE: Fully floating type, with hypoid pinion and banjo casing. Ratio, with standard axle, 6·66:1; with optional extra, spiral bevel, two-speed axle, 5·625:1 and 7·824:1.
SUSPENSION: Through axle, and semi-elliptic springs front and rear.
SHOCK ABSORBERS: $1\frac{7}{8}$ in diameter, direct acting units may be fitted at the front as optional extras.
STEERING: Marles, cam and double roller. Ratio 24:1. $4\frac{1}{4}$ turns from lock to lock. Turning circle 55 ft.
BRAKES: Girling 2LS front, and high centre lift 2LS type with external cylinders at the rear. Mechanical hand brake operates rear brakes. Vacuum servo assisted foot brake control. Drum diameter, front 16 in, rear $15\frac{1}{4}$ in. Shoe width, front 3 in, rear $4\frac{1}{4}$ in. Shoe area, front 184·4 in^2, rear 253·2 in^2.
TYRES: 9·00 × 20·00, 12 ply. Pressure, front 75 lb, rear 90 lb. Pierced steel disc type wheel with B6.5 × 20 wide base rims, 5·6 in offset.
DIMENSIONS: Wheelbase 14 ft $3\frac{1}{4}$ in. Track, front 5 ft 6 in, rear 5 ft $7\frac{1}{2}$ in. Ground clearance 10·68 in. Overall length 23 ft $6\frac{1}{4}$ in, overall width 7 ft $4\frac{13}{16}$ in over the rear tyres, overall height 7 ft $5\frac{3}{8}$ in, unladen. Height unladen of top of frame from ground, 2 ft $9\frac{3}{4}$ in at rear of chassis. Dry weight 6,542 lb. Engine dry weight 914 lb. Cab weight, white, 658 lb. Gearbox and clutch weight 229 lb.

Clutch and gearbox

A Borg and Beck, 13 in diameter single dry plate clutch with a spring



The cab of the Dodge 7 ton truck is carried on six rubber mountings on the frame

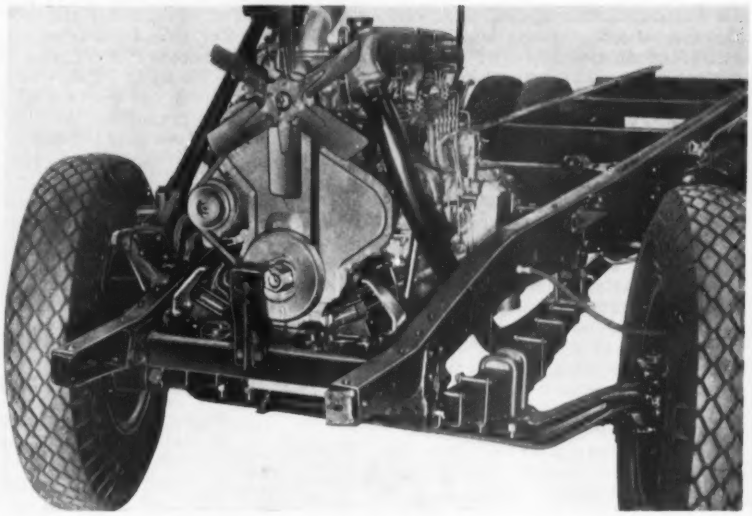


ARRANGEMENT OF THE DODGE 7 TON TRUCK CHASSIS
Wheelbase 14 ft 3½ in. Track, front 5 ft 6 in, rear 5 ft. 7¼ in

centre is employed. It has woven asbestos linings on Borglite segments, the friction lining area being 152 in^2. Sixteen pressure springs are fitted and their compression load when assembled is 2,640 lb. A ball thrust bearing is employed in the clutch actuating mechanism, and the whole unit is enclosed in a B.S.1452 grade 17, cast iron bellhousing bolted to the front face of the gearbox.

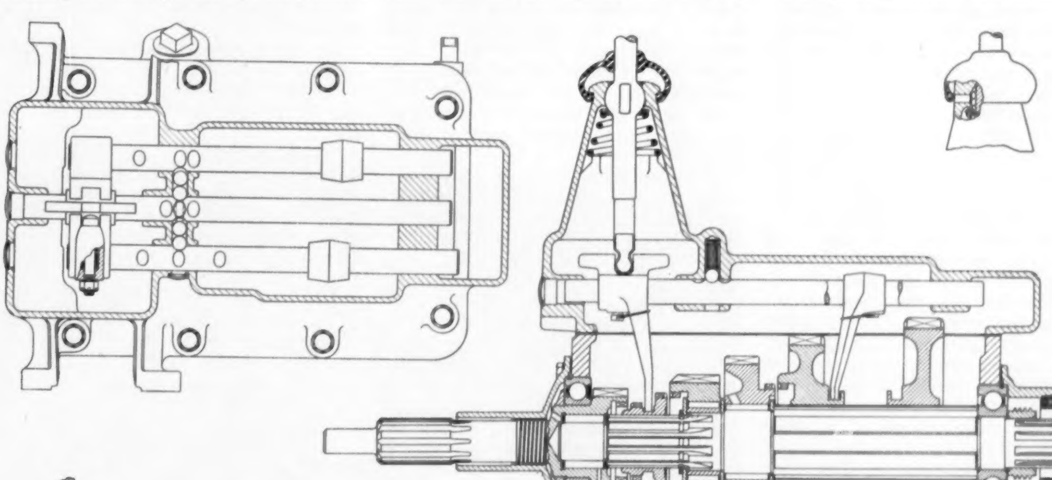
The dry weight of the gearbox is 249 lb with, and 188 lb without, the bellhousing. Five forward speeds and one reverse are obtainable, and all are of the crash type. Many operators consider that synchromesh gears are undesirable in commercial vehicles. There are a number of reasons for this: firstly, it reduces the cost of the unit, second, it increases reliability and reduces the amount of servicing needed, and third, the drivers of this class of vehicle generally are extremely experienced and have the skill necessary to effect gear changes without the slightest difficulty. The primary, fourth and third speed helical gears are of the constant mesh type and are engaged by means of toothed clutches. Second gear, and the combined first and reverse gear are of the straight spur type, and slide on the mainshaft to engage the appropriate layshaft and idler gears.

A B.S.1452 grade 17 casting forms the gearbox casing. The front and rear covers, which retain both the mainshaft bearings and also the rear bearing of the layshaft, are of cast iron, while the covers on each side of the box and the domed cap over the front bearing of the layshaft are of pressed steel. Another B.S.1452 grade 17 casting forms the top cover and houses the selector rods and striker mechanism. The unit holds 9 pints of oil, and S.A.E. 90 grade is recommended.


Rubber-to-metal bonded, double sandwich units are employed for the front mountings of the engine

The En 36 primary shaft is integral with the primary gear. Its front end, where it spigots into the tail end of the crankshaft, is $\frac{3}{4}$ in diameter, while it is $1\frac{1}{2}$ in diameter over the splines for the clutch centre member, and the spline depth is 0·15 in. Immediately to the rear of the splines is a $1\frac{3}{8}$ in diameter portion, and behind this, an oil return scroll is machined on the shaft. In cross section, this scroll is similar to a buttress-type thread and it works in an extension of the front cover, or mainshaft bearing retainer. Its diametral clearance is 0·006 in to 0·012 in.

A snap ring in a groove round the outer race of the ball bearing supporting the primary shaft is clamped between this front cover and the front wall of the gearbox. The inner race of



The gearbox is a simple five speed unit with sliding second, and bottom and reverse gears

this bearing is carried on a 55 mm diameter, shouldered portion of the shaft immediately in front of the primary gear, and is retained by a ring nut tightened against its front face. This bearing is lubricated partly by splash from inside the box and partly by oil centrifuged out of the mainshaft spigot bearing which is in a counterbore in its rear end. Two, $\frac{5}{32}$ in diameter holes are drilled to carry the oil from inside the counterbore to the front face of the shoulder immediately in front of the ring nut. From here, it is flung outwards, between the ring nut and front cover, to the bearing.

The mainshaft front spigot bearing is of the needle roller type. Its outer race is formed by the $1\frac{3}{8}$ in diameter counterbore in the primary gear while its inner race is formed by the $1\frac{5}{32}$ in diameter front end of the En 34 shaft. The rollers are of 0·95-1·1 per cent carbon steel. This bearing is lubricated by oil forced through two $\frac{1}{8}$ in diameter holes drilled radially from the roots of the gear teeth into the counterbore.

Supporting the other end of the shaft is a ball bearing housed in the rear wall of the gearbox. The outer race of this bearing is located axially by a snap ring in a groove round its periphery. This ring is clamped between the bolted on cover and the gearbox end wall. A rearward extension of the cover houses a leather oil seal, of the spring-loaded lip type, which bears on the boss of the companion flange for the universal joint. This flange is splined on to the shaft. The outside diameter of the splined portion of the shaft is $1\frac{1}{2}$ in and the spline depth is 0·08 in. A slotted nut on the 1 in diameter tail end of the shaft pulls the flange boss against the speedo gear. The front face of this gear bears against the inner race of the ball bearing, and clamps it against a shoulder in front of the 40 mm diameter portion of the shaft, on which it is carried. Thus, axial location of the mainshaft assembly is effected by this rear bearing.

Immediately in front of the rear bearing, the $2\frac{1}{4}$ in diameter mainshaft is splined to a depth of $\frac{1}{8}$ in. Sliding on these splines are the combined reverse and first speed gear, and the second speed gear. The groove for the selector fork for the first and reverse speed gear is formed round the front end of the gear boss, while that for the second speed gear is at the rear of its boss. A forward extension of the second speed gear forms the outer member of a toothed clutch, the pitch circle diameter of which is 3·428 in. When the gear is moved forwards by the selector fork, the outer member of the clutch engages the inner member, which is formed on the rearward extension of the third speed gear. On the other hand, when it is slid to the rear, the gear engages the second speed pinion on the layshaft. The ends of the gear and pinion teeth are rounded to facilitate engagement.

The mainshaft third speed gear runs on a $1\frac{5}{16}$ in long needle roller bearing; the outer race is formed by the $2\frac{5}{16}$ in diameter bore in the gear boss and the inner race is the shaft, which at this point is $1\frac{29}{32}$ in diameter. A $\frac{1}{8}$ in thick, En 33 thrust washer is housed in a $\frac{1}{8}$ in deep annular recess in the rear end of the gear boss and interposed between the gear and the splined portion of the shaft. Another washer of the same material and thickness is interposed between the ends of the third and fourth speed gears. This washer is assembled on to the shaft from the front and bears against a shoulder at the front of the journal for the third speed gear.

A needle roller bearing, $1\frac{5}{8}$ in long, carries the fourth speed gear. However, in this case, although the outer race is formed by the $2\frac{7}{16}$ in diameter bore of the gear boss, the inner race is a separate, case hardened, En 34 sleeve round the $1\frac{3}{4}$ in diameter shaft. This sleeve is fitted so that the splines on the shaft may run out inside it without interfering with the bearing. Its outside diameter is $2\frac{1}{32}$ in and it is located against rotation by a shouldered dowel. The larger diameter end of the dowel registers between two splines on the shaft and the smaller diameter portion is in a hole in the bush. Therefore, it cannot slide into the bush and jam the bearing. The whole assembly is retained by another $\frac{1}{8}$ in thick thrust washer and a snap ring in a groove round the splined front end of the mainshaft.

Sliding on these splines, which are 1·604 in outside diameter by $\frac{1}{8}$ in deep, is the En 34 inner member of another toothed clutch. Two rows of teeth are machined on the En 34 inner member of this clutch, and between them is the channel for the selector fork. The row at the rear has a pitch circle diameter of 2·857 in and, when fourth speed is selected, it engages teeth machined in a forward extension of the fourth speed gear; that at the front has a pitch circle diameter of 2·285 in and, when it is slid into another toothed outer member formed in the rear end of the primary gear, transmits the direct top speed drive. On all clutches, the leading ends of the teeth are rounded to facilitate engagement, and all except three teeth grouped together are relieved, by 0·014-0·016 in, on the face opposite to the thrust side. This obviates the necessity of having to maintain close tolerances on the spacing of the teeth, except the unrelieved ones, and facilitates engagement.

GEARBOX DATA

	Pitch circle diameter	Diametral pitch	Gear thickness	Material
Primary gear	3 in	7/9	$1\frac{1}{16}$ in	En 36
Mainshaft gears;				
4th speed	3·857 in	7/9	$1\frac{1}{2}$ in	En 34
3rd speed	4·964 in	7/9	$1\frac{1}{16}$ in	En 34
2nd speed	6·313 in	6/8	$1\frac{1}{16}$ in	En 34
1st and reverse speeds	7·298 in	6/8	$1\frac{1}{4}$ in	En 34
Layshaft gears;				
Layshaft gear	6·5 in	7/9	$1\frac{1}{16}$ in	En 34
4th speed	5·643 in	7/9	$1\frac{1}{16}$ in	En 34
3rd speed	4·536 in	7/9	$1\frac{1}{16}$ in	En 34
2nd speed	3·186 in	6/8	$1\frac{1}{16}$ in	En 36
1st speed	2·02 in	6/8	$1\frac{1}{16}$ in	En 36
Reverse	2·55 in	6/8	$1\frac{1}{16}$ in	En 36
Reverse idlers	3·95 in and 3·53 in	6/8	1 in	En 34

The En 36 layshaft is 2 in diameter at the front where the layshaft gear, and fourth and third speed gears are keyed on. These three gears are retained by a circlip in a groove immediately in front of the layshaft gear. At the centre, where the reverse and second speed gears are formed integrally with the shaft, the diameter is stepped up to $2\frac{1}{32}$ in diameter. Behind these two gears, the shaft diameter is stepped down to $1\frac{7}{8}$ in where the bottom speed gear is also integral with the shaft.

At the extreme front end, the layshaft is carried in a roller bearing. The outer race is pressed into its housing in the front wall and the inner race is formed by the shaft, which at this point is turned down to $1\frac{23}{32}$ in diameter. A domed, pressed-steel cap seals the outer end of the bearing housing in which is machined a groove for a circlip that retains the cap. Immediately behind the bearing is a chip-shield pressed on the shaft.

The rear end of the shaft is turned down to $1\frac{9}{16}$ in diameter to carry the inner race of a ball bearing. This bearing is housed in the end wall of the gearbox, and its outer race is retained by a snap ring in a groove round its outer periphery. The snap ring is clamped between a bolted-on rear cover and a split, steel spacer-ring interposed between it and the wall of the gearbox. The inner race of the bearing is retained by a $\frac{1}{4}$ in thick plate pulled against its rear face by two $\frac{3}{8}$ in diameter set bolts screwed into the end of the shaft. Wire is used to lock the two bolts together, although the tendency for them to work loose is not so great as it might be if only one bolt, screwed axially into the shaft, were employed. All the teeth of the helical gears are at a helix angle of 23 deg 12 min 57 sec. Other data concerning the gears is given in the

accompanying table.

A $1\frac{1}{8}$ in diameter, En 32 spindle carries the two, reverse idler gears. These gears are hobbed one at each end of a sleeve which is carried on needle roller bearings running directly on the spindle. A tubular distance piece separates the bearings, the centres of which are approximately $4\frac{1}{2}$ in apart. The rear end of the spindle is carried in the end wall of the gearbox, to which is bolted a locating plate that registers in a groove machined chordwise in the periphery of the spindle. The front end of the spindle is spigoted into a boss on the side wall of the box. Engagement of reverse gear is effected by sliding the mainshaft gear to engage the idler gear at the rear end of the sleeve. The gear at the front end is in constant mesh with another on the layshaft.

Three En 32 selector rods are positioned side by side in the top cover of the gearbox. Each rod is carried in three bosses, the intermediate one housing the gear lock and interlock devices. The gear lock is a simple spring-loaded ball arrangement, seating in a circular section, transverse groove machined chordwise on the periphery of each rod. An interlock mechanism of conventional form is employed. Two balls on each side of the centre rod are positioned with their centres in line with the axis of a short pin in a hole drilled diametrically through the centre rod. The balls each register in a groove in the adjacent rod. When one of the rods is moved to select a gear, it unseats the ball that registers in its groove and forces the other balls and the pin sideways to register in grooves in the other two rods, thus locking them in position.

The selector and striker forks are of En 8. They are assembled on to the rods and secured by set screws carried radially in their bosses and locked by wire. A spring loaded plunger operating in the striker fork for bottom and reverse gear forms the baulk. It is not necessary under normal driving conditions to use bottom gear, which is employed only in unusually difficult circumstances, such as starting on exceptionally steep hills.

Back axle

A Hardy Spicer propeller shaft transmits the drive to the rear axle. It is supported near the centre by a self-aligning, double-row ball bearing, bolted directly to one of the intermediate cross members of the frame. The front portion of the shaft is $2\frac{1}{4}$ in diameter and $54\frac{27}{32}$ in long, while the dimensions of the rear portion are 3 in diameter by $62\frac{13}{16}$ in long. Needle roller universal couplings are employed throughout and the sliding joint is incorporated at the front end of the rear portion of the shaft.

The rear axle is of the fully-floating type with a hypoid pinion, and is enclosed in a banjo-type casing. The final drive ratio is 6·66 : 1. This axle, complete with wheels, tyres and brakes, weighs 764 lb dry. Operators who require a two-speed axle may have an Eaton 16,500 unit which is supplied as an optional extra. Its final drive ratios are 5·625 : 1 and 7·824 : 1, and it weighs 931 lb dry.

In the standard, single speed axle, the gear carrier is of malleable cast iron. Sixteen, $\frac{5}{8}$ in diameter B.S.F. set-bolts secure the banjo casing to the carrier, and a pressed steel cover is welded on at the back. The oil capacity of the unit is 10 pints, and S.A.E. 90 grade hypoid gear oil is specified.

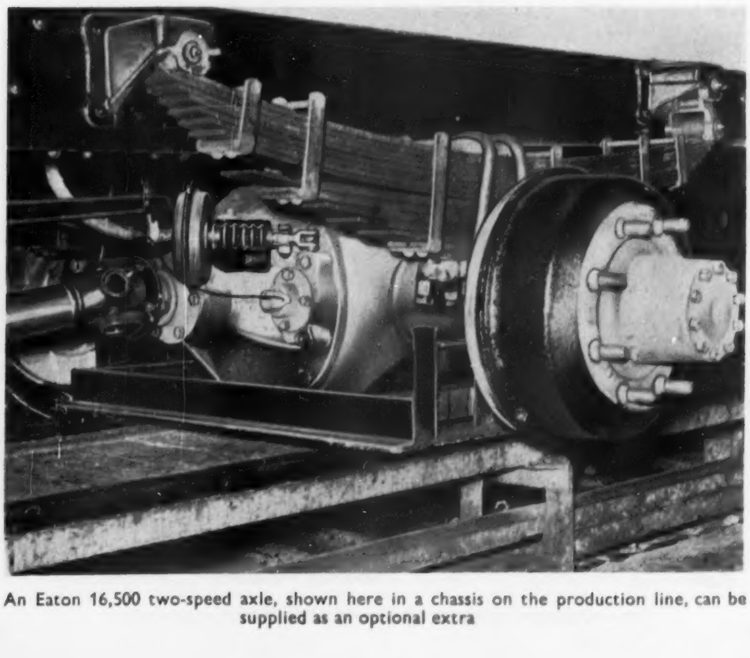
There are six teeth on the crown wheel pinion, which is integral with its En 36 shaft. This shaft is straddle mounted between two opposed taper roller bearings at the front, and one plain roller bearing at the rear. Its axis is offset $1\frac{1}{2}$ in below that of the crown wheel, and $\frac{1}{4}$ in to the right-hand side of the plane in which are the axes of the differential pinion spider arms. The shaft diameter, where it carries the taper roller bearings, is 2 in and the bearing centres are approximately $1\frac{3}{4}$ in apart. Their outer races are carried in a separate steel housing into which they are assembled from each end. This housing is spigoted into the nose of the gear carrier casting and secured, together with a malleable iron end plate, by six $\frac{7}{16}$ in diameter set bolts. A DA type oil seal is housed in the end plate, which is spigoted into the front end of the bearing housing.

This oil seal bears on the boss of the universal joint flange which is splined on the front end of the shaft. The whole assembly, comprising the companion flange, the roller bearings and the $\frac{3}{4}$ in long distance tube which separates their inner races, is pulled against the hypoid pinion by a slotted nut and washer on the end of the shaft. Adjustment to the axial position of the pinion is effected by means of shims interposed between the nose of the gear carrier and the flange round the bearing housing. The bearing pre-load is controlled by shims between the tubular distance piece and the inner race of the front taper roller bearing.

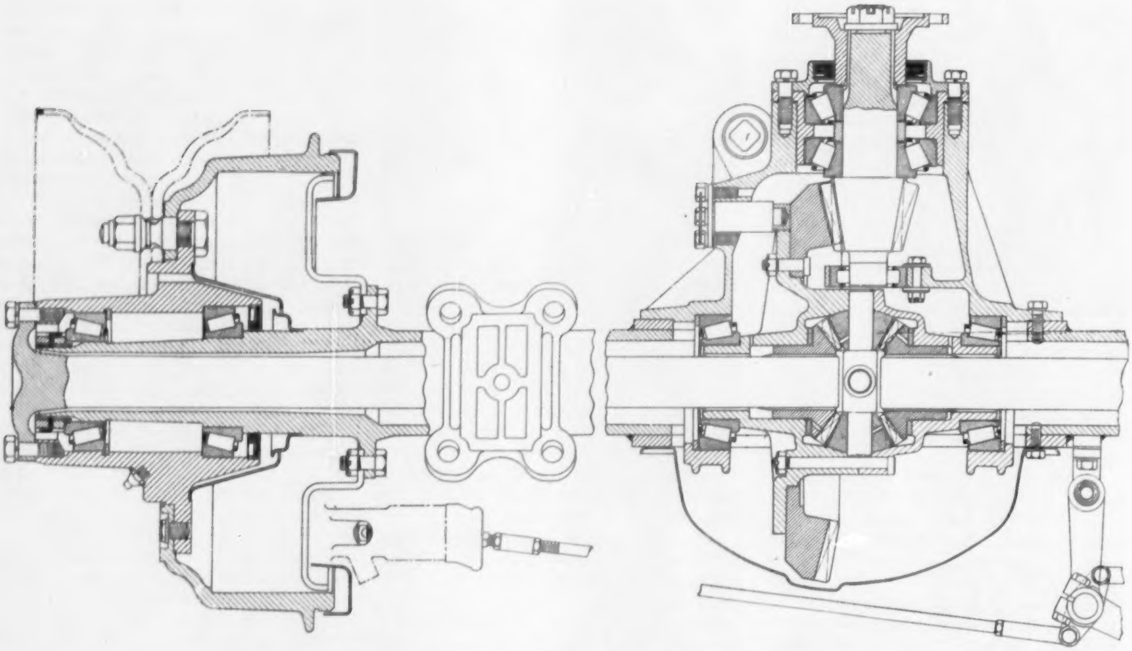
The outer race of the support bearing at the rear end of the shaft is housed in a boss cast in the gear carrier. It is retained by a $\frac{7}{16}$ in diameter bolt and nut and two plain washers. This bolt is passed through the bearing housing boss, and pulls the washers, which are of such a size that they extend over the periphery of the housing, against the front and rear faces of the outer race of the bearing. The rollers run directly on the front extension of the pinion shaft which is ground to 1·416 in diameter.

An En 36 stamping forms the crown wheel, which is spigoted on to the inner face of a flange formed round the cage and secured by twelve $\frac{7}{16}$ in diameter, special bolts. To one side of the pinion, the crown wheel is supported by a phosphor bronze thrust pad. This pad is of thimble shape and is mounted on the $\frac{5}{8}$ in diameter shouldered end of a $1\frac{1}{8}$ in diameter strut which is a push fit in a hole in the boss on the side of the gear carrier. Round the outer end of the strut is a $\frac{5}{16}$ in thick flange drilled for the four $\frac{3}{8}$ in diameter set-bolts which secure it to the gear carrier. Between the flange and the gear carrier are shims by means of which the axial location of the thrust pad assembly is adjusted.

Two 40 ton, carbon steel stampings are bolted together to form the differential cage. It is carried in two taper roller bearings, one on each side, spaced with their centres approximately $9\frac{3}{4}$ in apart. Registering against the outer face of each of these caps are two projections on the gear carrier casting. The inner races bear against shoulders on



An Eaton 16,500 two-speed axle, shown here in a chassis on the production line, can be supplied as an optional extra



A straddle-mounted hypoid pinion is employed in the single-speed rear axle

the cage and the outer races are retained in their housings in the gear carrier by bolted-on caps. The bearing pre-load and the mesh of the crown wheel and pinion are adjusted by means of distance washers between the outer races of the bearings and shoulders in their housings.

Inside the cage, four $2\frac{3}{4}$ in diameter differential pinions are carried on the $\frac{7}{8}$ in diameter arms of a $4\frac{1}{2}$ per cent nickel steel spider. The pinions are of $3\frac{1}{2}$ per cent nickel chrome steel, and are case hardened. Their spherical outer faces bear directly in the cage and three drillings from the roots of their teeth feed oil into their journal bearing surfaces.

The $4\frac{3}{16}$ in diameter differential gears are splined on to the ends of the half shafts and their $2\frac{5}{8}$ in diameter bosses bear in the differential cage for a length of about $1\frac{3}{16}$ in. Interposed between the outer face of each gear and the cage is a $3\frac{7}{8}$ in diameter by $\frac{3}{16}$ in thick, phosphor bronze thrust washer. The length of tooth engagement between the differential pinions and gears is $1\frac{1}{8}$ in.

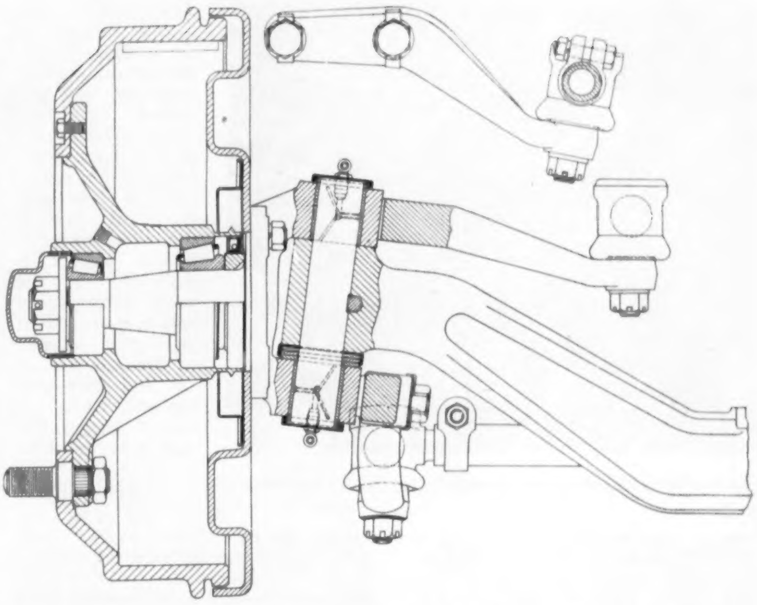
Raised on the periphery at the inner end of each En 24 stamped half shaft are the splines that carry the differential gears. Their outside diameter is 2 in, and the root diameter is $1\frac{7}{8}$ in. At the outer end of the shaft, the driving flange is upset and secured to the hub by eight $\frac{1}{2}$ in diameter set bolts. The axle casing is divided at the centre to form the banjo. At the points where it divides it is of rectangular section, $3\frac{3}{4}$ in wide by $4\frac{1}{4}$ in deep, and its wall thickness is $\frac{3}{8}$ in. At its outer end, its overall cross sectional dimensions are $3\frac{3}{4}$ in by $3\frac{3}{4}$ in.

A 0·45 per cent carbon steel stub axle, on the inner end of which is upended the brake back-plate carrier flange, is projection welded to the axle casing. The hub assembly is carried on two taper roller bearings on the stub axle and is secured by two ring nuts between which is a tab washer. One nut is tightened against the inner race of the outer bearing and the other nut, together with the tab washer, is used solely for locking purposes. The outer race is pulled against a snap ring in the bore of the 0·3 per cent carbon steel hub in which it is housed. A domed pressed steel cap in the outer end of the hub retains the grease and prevents the ingress of foreign matter. The outer race of the inner roller bearing is assembled into the hub from the other end and its inner race is pulled against a shoulder on the stub axle. Also carried in the inner end of the hub, and bearing on the stub axle which is $3\frac{11}{16}$ in outside diameter at this point, is a leather oil seal of the spring-loaded lip type. At the inner bearing, the outside diameter of the stub axle is $3\frac{1}{4}$ in while at the outer one it is $2\frac{7}{8}$ in. Its inside diameter is $2\frac{1}{8}$ in.

Immediately inboard of the oil seal,

Arrangement of the single-speed axle and rear suspension

a cupped thrower ring is pressed on to the stub axle. It is shrouded by a pierced, cupped pressing spigoted into the hub, which is peened over the edge of the shroud to retain it. This pressing forms the trap from which any lubricant that may pass the seal is drained away through drillings in the hub flange. Chromium alloy, cast iron brake drums are spigoted on to the hub flange and secured by countersunk set bolts. Eight $\frac{7}{8}$ in diameter, alloy steel studs, together with spherical seating nuts, secure the wheels. These studs have collars formed round them approximately mid-way between their ends, and their shanks are serrated immediately inboard of the collar to locate them against rotation in the holes in the hub flange into which they are pressed. The collars are pulled against the flange by nuts on the inner ends of the studs, and they locate the brake drum. Their front faces are of conical shape to form a seating for the wheel disc.



A three-washer type bearing between the swivel pin knuckles takes the thrust

Suspension

At the rear, conventional two-stage, semi-elliptic springs are employed. They are 54 in long and are of silicon-manganese-steel. The primary section of each spring has eight leaves, $\frac{3}{8}$ in thick, while the secondary section has seven leaves, $\frac{9}{16}$ in thick. All are 3 in wide. At each end of the spring, the second leaf is extended partly round the eye so that, if the top leaf should break, there is less chance of the whole unit collapsing. The bump stops are in the form of rubber blocks underneath the frame side members, and they bear directly on the axle when it moves to the full bump position.

To the fully laden position, the deflection is 4·115 in and to full bump it is 6·865 in. Up to a load of 1,600 lb, the spring rate is 2·58 in/ton. This gives a normal periodicity of about 92·6 cycles per minute. At higher loads, when the secondary spring comes into operation, the rate is 0·767 in/ton. The unsprung weight of the rear axle with the springs, and four wheels and tyres, is 1,996 lb.

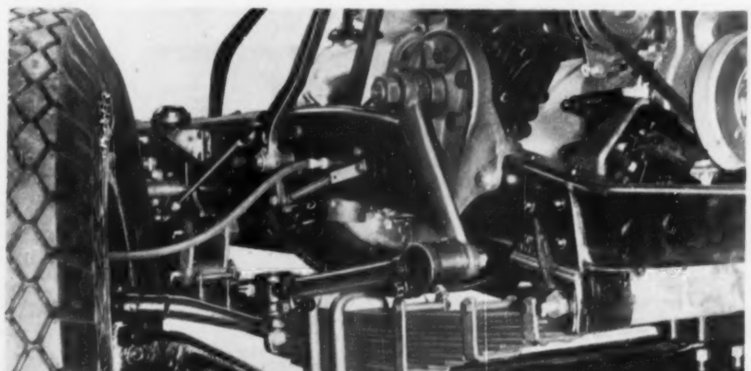
The front eye of the rear spring and the shackle at the back are carried in malleable iron castings riveted on to the outer face of the frame side members where they are supported by cross members. The pins are $1\frac{1}{8}$ in diameter and are of a steel containing 0·15 per cent carbon, 0·3 per cent silicon, 0·5 per cent manganese, 0·07 per cent sulphur and 0·07 per cent phosphorus. At the centre, the spring is mounted on stamped En 3 plate welded on top of the axle. It is held down by two $\frac{7}{8}$ in diameter U-bolts, of 50-55 ton steel.

For the front suspension a conventional Elliott type, through axle and semi-elliptic spring arrangement has been adopted. Direct acting, $1\frac{3}{8}$ in diameter by $6\frac{1}{2}$ in stroke, shock absorbers may be fitted, if required, by the customer. The swivel pin angle is 6 deg and the castor and camber angles are each $1\frac{1}{2}$ deg. A toe-in of $\frac{1}{8}$ in to $\frac{3}{16}$ in is specified. The unsprung weight of the axle, springs, wheels and tyres is 1,015 lb. To the fully laden position, the wheel deflection is 2·54 in, while to full bump it is 5·665 in. The front end rate is 2·083 in/ton. This gives a periodicity of 118 cycles per minute.

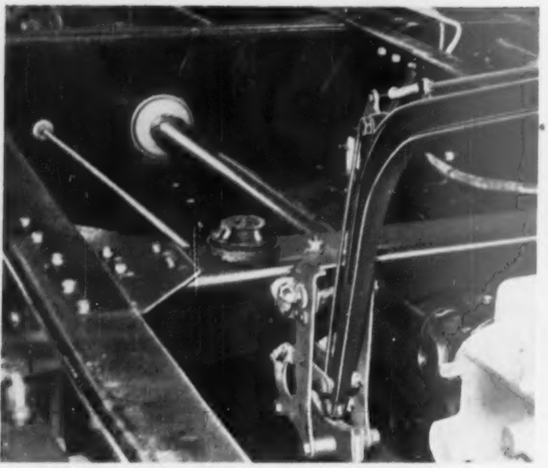
Silicon-manganese-steel springs are employed, and their length between eye centres is 45 in. Each is $2\frac{1}{2}$ in wide and comprises fourteen leaves, $\frac{5}{16}$ in thick. The eye at the front and the shackle at the rear are carried on 1 in diameter pins in malleable iron castings riveted to the frame side member. These pins are of the same material as those employed on the rear suspension. Between the front pair of castings, the frame is well supported by the front cross member. However, it has not been possible to place the cross member immediately between the shackle brackets; instead, it is positioned approximately 6 in further forwards, and passes under the engine. The bump stops are blocks of rubber, which are mounted under the side frames and which bear on the spring saddles.

An I-section, En 16 axle is employed. At the centre, it is approximately $3\frac{1}{8}$ in deep by $2\frac{1}{2}$ in over the flanges. The web thickness is $\frac{1}{2}$ in, and the flanges also are approximately $\frac{1}{2}$ in thick. Immediately outboard of the spring seating pads, which are integral with the remainder of the forging, the axle is upswept, approximately 5 in, to the boss that carries the swivel pin.

The swivel pin is of Ubas, bright steel bar, 1·390 in diameter. Machined on its periphery, approximately midway between its two ends, is a flat against which is registered the $\frac{1}{2}$ in diameter cotter pin that is carried in the boss to locate it. The length of the



Simple, semi-elliptic springs are employed for the front suspension



The bracket for the hand brake is mounted on a frame cross member

boss is approximately 3¼ in, and the Clevite bushes in the two knuckles, one above and the other below it, are each 1⅞ in long. A Compo, three-washer thrust bearing is interposed between the bottom face of the boss on the axle and the lower knuckle, and a cupped steel shroud is fitted over the thrust bearing to protect it from foreign matter. The outer pair of thrust washers is of manganese bronze and the inner one is of case hardened mild steel.

A 5/32 in diameter axial hole is drilled about 1¼ in deep in each end of the swivel pin. The outer ends of these holes are counterbored and tapped to receive grease nipples, which are passed through holes in the centres of cupped steel retainers for the rubber rings that form the grease seals. These rings seat round the ends of the bushes, which project about ⅛ in beyond the outer faces of the knuckles, and the nipples pull the retainers against the projecting ends of the swivel pin. From the axial drilling at each end, grease passes through a radial hole to a groove extending about half-way round the bearing surface, mid-way between the ends of the bush. One end of the groove is turned upwards and the other downwards to spread the grease over the whole length of the bearing. The upward turned end of the groove in the lower bearing breaks out at the end of the bush to pass the lubricant to the thrust washers. This feature is retained in the top bearing for the sake of interchangeability.

An En 16T stub axle knuckle forging is employed and the wheel is carried on two taper roller bearings. At the inner bearing, the stub axle is 2 in diameter while at the outer one it is 1½ in diameter. The bearings are assembled into each end of the malleable cast iron hub, in which there are shoulders to locate the outer races. The whole assembly is secured by means of a slotted nut on the 1 in diameter threaded end of the stub axle. This nut is tightened against a ¼ in thick washer, which is of such a diameter that should failure of the bearings occur, the wheel cannot come off, and this washer is pulled against the inner race of the outer bearing. At the opposite end of the hub, the inner race of the other bearing is pulled against a distance ring, the inner periphery of which is shaped so as to clear the fillet at the junction of the stub axle with the brake back plate carrier flange. The radius of the fillet is ¼ in.

A leather oil seal, of the spring-loaded lip type, bears on the outer periphery of the distance ring, and is housed in an extension of the hub. Round this extension is formed a collar of triangular section, to act as a thrower. Should any lubricant pass the seal, it is flung off the thrower and into a dished steel trap bolted to the brake back plate. This back plate is spigoted and bolted to its ½ in thick, carrier flange round the stub axle forging. The cast iron brake drum is spigoted on a flange round the hub and secured in a manner similar to that described for the rear axle.

Steering and brakes

Marles, double-cam-and-roller type steering gear is employed. It is bolted to the frame at a point adjacent to the end of the front cross member. The ratio is 24:1, and gives 4½ turns of the 20 in diameter wheel from lock to lock. On full lock, the angle of the inner wheel is 40 deg while that of the outer one is 31 deg. This gives a turning circle of 55 ft.

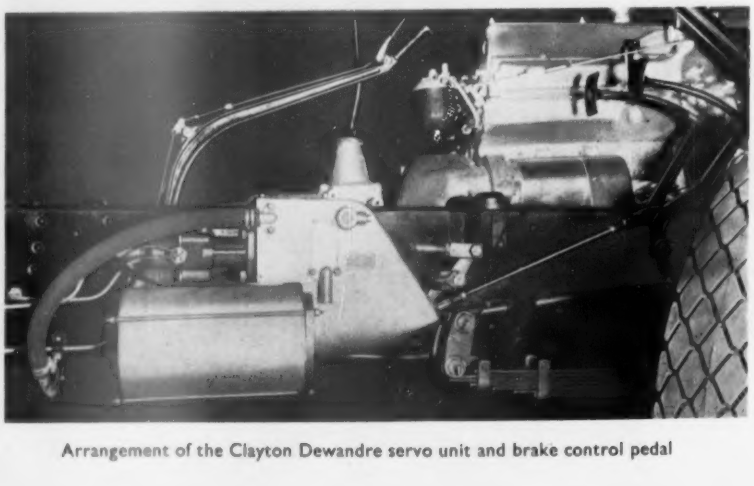
A conventional steering layout, comprising a drag link to the steering arm on one wheel, and a track rod, has been adopted. A stamped En 16 drop arm is employed and it has an effective length of 9 in. The drag link is 1¼ in diameter by 6 S.W.G. tube, and its centre-to-centre length is 15 in. An En 16T steering arm, which has an effective length of 9 in, is secured by two, ⅝ in diameter B.S.F. bolts spaced 3 9/16 in apart on each side of the top knuckle on the stub axle forging.

The 7½ in long, En 16T track rod arm at each wheel is secured by two bolts to the lower knuckle of the stub axle forging. These bolts are ⅝ in diameter and their axes are spaced 3⅜ in apart. The track rod is 1¼ in diameter by 6 S.W.G. steel tube and its centre-to-centre length is 56½ in. All the end fittings used on the drag link and track rod are 1¼ in diameter ball joints of the self adjusting type screwed into the split ends of the tube. After adjustment has been effected, clamping rings are tightened round the split ends of the tubes by ⅜ in diameter bolts and nuts.

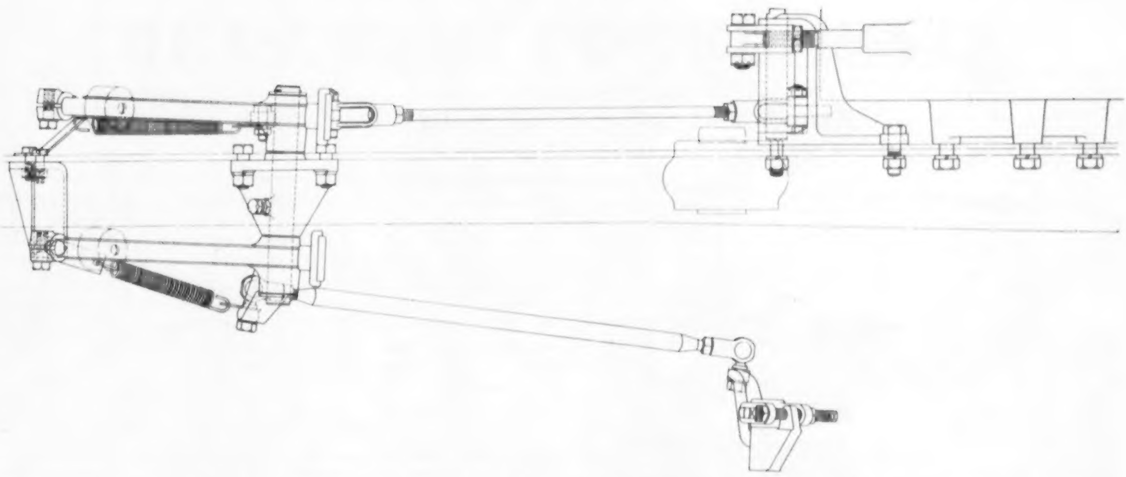
Girling brakes are employed. Two leading shoe units are fitted both at the front and at the rear. Whereas the brakes at the front have internal cylinders, those at the back are of the high-centre lift type, and have external cylinders. Their drum diameters are 16 in and 15¼ in respectively. The width of the front brake shoes is 3 in while that of the rear is 4¼ in, and the shoe lining areas are respectively 184·4 in^2 and 253 in^2 per brake.

The brake pedal is carried on a 2 in long, Oilite bush at one end of the ⅞ in diameter, En 6, bright mild steel pivot pin. On the other end of the pin is the clutch pedal, also on an Oilite bush; and between the two is a cast boss, bolted to the frame side member. The pin is assembled into this boss and locked by a dowel ended bolt screwed radially into the boss and registering in a hole in the pin. Each pedal is retained by a split pin and washer at the adjacent end of the pin.

A brake pedal lever-ratio of 4:1 has been adopted, and the length of travel is 8 in. This pedal is connected by means of a ⅜ in diameter rod, with adjustable fork-ends, to one of two eyes on a pendant lever on a short, tubular cross shaft. The two eyes are



Arrangement of the Clayton Dewandre servo unit and brake control pedal



To allow for the relatively large amplitude of motion of the engine on its mountings, roller socket end fittings are employed at each end of the rod connected to the clutch pedal

incorporated to make the lever common to both left and right hand drive vehicles. The cross shaft is $1\frac{5}{16}$ in outside diameter by $\frac{3}{4}$ in inside diameter and $3\frac{13}{16}$ in long, and is carried, on a pin, between the lugs of a malleable iron bracket bolted to the frame side member. A plain lever, which is upstanding at the other end of this shaft, is connected by means of an adjustable fork-end to the Clayton Dewandre, 175 mm bore, brake servo unit.

A conventional hand brake lever is mounted on a bracket on a frame cross member immediately to the rear of the gearbox. Its lever-ratio is 7·32 : 1. A fork end, screwed on to a $\frac{3}{8}$ in diameter rod, is pinned to the lower end of the lever. At the centre of the rod, support is afforded by an Oilite bush in a rubber grommeted hole in the cross member carrying the intermediate bearing of the propeller shaft. The function of the grommet is to allow for vertical motion arising from the action of the pendant lever that supports the rear end of the rod.

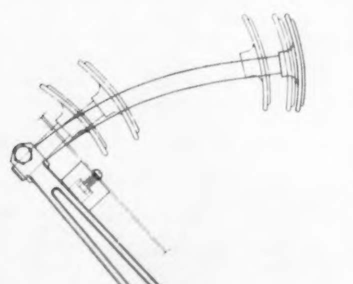
This lever is pivoted on a bracket on the cross member immediately in front of the axle, and it also carries another rod connecting it to the pivoted and swinging link compensator, bolted to a bracket on the axle. The compensator is pivoted on Oilite bushes on a $\frac{3}{4}$ in diameter, bright drawn, mild steel pin.

Frame

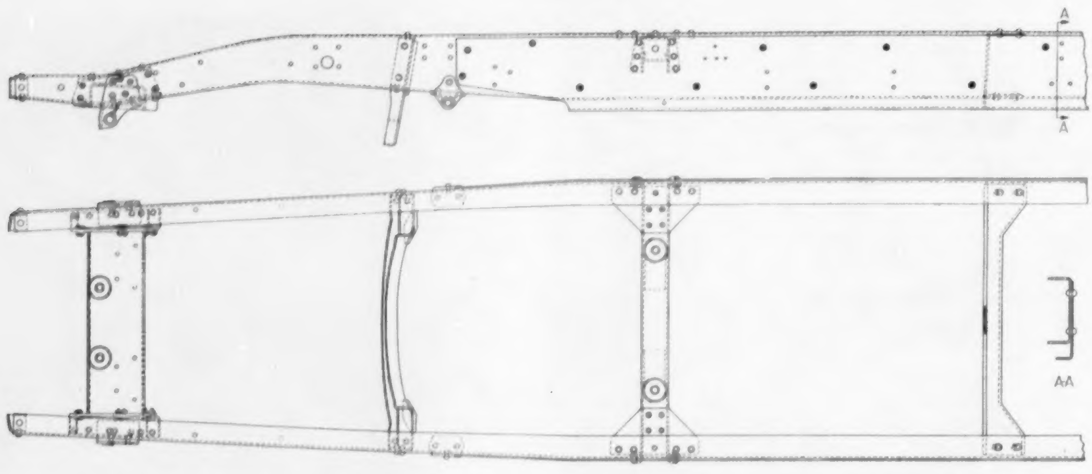
The frame consists of two channel section side members and six cross members, all of which are made of En 2A. As viewed from above, it tapers from $28\frac{33}{64}$ in wide at the front to $35\frac{19}{32}$ in wide at the rear. Wherever possible, cold riveting is employed at the joints, because with hot riveting there is a danger of the rivets becoming loose as a result of contraction on cooling. At the centre, the side members are reinforced by a flitch plate, the lower edge of which is flanged. This plate is $\frac{3}{16}$ in thick, and its flange is 2 in wide. The flitches are each secured to the web of their respective side member by two rows of $\frac{7}{16}$ in diameter rivets at approximately 16 in pitch, and their ends are scarfed. One row of rivets is about $1\frac{9}{16}$ in below the top flange and the other is the same distance above the bottom flange. At the centre, the channel-section side members are $\frac{7}{32}$ in thick by $8\frac{1}{2}$ in deep by 3 in wide over the flanges. The total depth over the flitch and channel is 10 in. At the extreme rear end the side members are $5\frac{1}{4}$ in deep, while at the front they are 3 in deep.

An inverted channel-section forms the front cross member, which is bolted on, so that it may be removed to facilitate engine replacement. The cross-sectional dimensions of this member are $\frac{7}{32}$ in thick by 3 in deep by $7\frac{3}{8}$ in wide. Welded to each end is a $\frac{3}{8}$ in thick steel plate, through which six $\frac{7}{16}$ in diameter B.S.F. bolts are passed to secure it to a bracket on the frame side member. This bracket is of channel-section, about 11 in long, and its maximum width is 3 in. Its flanges are riveted inside the flanges of the side member so as to form a box-section at this point.

The second cross member is of channel-section, $\frac{3}{16}$ in thick by $1\frac{1}{2}$ in wide by $3\frac{1}{4}$ in deep at the centre where it is swept downwards to pass under the engine. As has already been mentioned, it is not positioned directly between the brackets for the front spring shackles, but is about 6 in in front of them. The third cross member is of inverted channel-section, $\frac{3}{16}$ in thick by $4\frac{9}{16}$ in deep by $3\frac{1}{2}$ in wide. It



The motion of the brake pedal is transmitted to the servo unit by means of a tie rod and cross shaft



A bolted-on front cross member is employed so that it may be taken off to facilitate removal of the engine

is gusseted to the side members and, for a length of approximately 14 in from each end, it is closed by a welded-in channel-section of the same thickness. This cross member carries the rear pair of Firestone mountings for the cab. Another pair of mountings is carried on $\frac{5}{32}$ in thick outrigger brackets, at a point immediately above the rear spring shackle bracket on each frame side member. Two more are on the front cross member. Those at the front are spaced with their centres 9 in apart while those at the rear are 18 in apart.

A $\frac{7}{32}$ in thick, channel-section pressing forms the fourth cross member which supports the intermediate bearing of the propeller shaft. This member is $8\frac{1}{16}$ in deep and its width over the flanges is 5 in at each end and 2 in at the centre. The fifth cross member supports the frame side members where they carry the brackets for the spring eyes of the rear suspension. This member is of inverted channel-section and, between the gussets at each end, it has a closing plate welded to the lower edges of its flanges. It is $\frac{5}{32}$ in thick by $3\frac{1}{2}$ in deep by $3\frac{1}{2}$ in wide. The upper gussets are flat plates, but the lower ones are of top hat section, the flanges of which form the gusset plates. Their inner ends are welded inside the cross member, and their outer ends to the bottom flanges of the side members. At the rear, between the shackle brackets, is a channel-section cross member, $\frac{5}{32}$ in thick by $6\frac{13}{32}$ in deep. Its top flange is $1\frac{5}{8}$ in wide and its bottom flange is 4 in wide.

Electrical equipment

Two 12-volt batteries connected in series are employed to supply the 24-volt system. Each has a capacity of 81 amp-hr at a 10 hr rate. They are served by a C.A.V. G524-9 dynamo, used in conjunction with a C.A.V. 203-1 voltage regulator. The head lamps are of the flush fitting F700, 24-volt type, and the side lamps are LD 109A units. Two stop and tail lamps are fitted and they are the BR 1205/B type. HF 1235 horns are employed. The system is protected by an SF6 fuse box.

DUST AND ENGINE WEAR

IN an article entitled "The Cleaning of Engine Air. Dust and its Effect on Engine Wear," by J. L. Koffman, in *Gas and Oil Power*, Vol. 48, No. 572, the problem of engine air cleaning is considered in the light of the author's experiences with engines of fighting vehicles. Engine air cleaners are not required to deal with the whole wide range of the size of air impurities. Trapping particles the size of which exceeds 100μ presents little difficulty, and there is no reason to stop those which are so small that they cannot cause undue wear. Therefore, to draw up specifications for air cleaner performance and test dust, it is necessary to know the particle size of airborne dusts, their specific gravity, the dust concentration likely to be encountered by vehicles, and particle size range and dust concentration likely to harm the engine.

Methods of sampling and analyzing dust are described. From the analysis of 200 samples, a specification has been laid down for test-dust for fighting vehicle air cleaners. Such test-dusts are obtained by blending ground silica powder from two mills. Whereas, under severe conditions, dust concentration can be as high as 40 mg/ft^3 of free air, concentrations exceeding 10 mg/ft^3 are rarely encountered except in dust storms.

Tests on four-cylinder engines of $3\frac{3}{8}$ in bore and $3\frac{1}{2}$ in stroke with pistons, rings and cylinder liners of grey iron show that dust sizes of 5-15μ produce 150 per cent as much wear as those of 0-5μ, and dust measuring 15-30μ gives 250 per cent more than that measuring 0-5μ. Dusts were fed at the rate of 25 mg/ft^3 of air. These tests showed that particles measuring even less than 5μ should be removed from the air, since the average rate of liner wear with 0-5μ dust is 0·00135 in in eight hours. Other test results confirm the desirability of eliminating particles down to about 2μ in size.

Under severe conditions, a six-cylinder diesel bus engine will aspirate some 40 mg of dust with each cubic foot of air, and even with a cleaner efficiency of 99 per cent, the rate of entry of dust to the engine is 5·3 g/hr. Therefore a minimum cleaner efficiency of 99 per cent is desirable for vehicles operating under dusty conditions. Under more normal conditions, an efficiency of 98 per cent may suffice, but the difference of 1 per cent involves doubling the amount of dust reaching the engine. *M.I.R.A. Abstract No.* 6320.

Giant Pneumatic Tyres

A NEW publication, entitled "The Care of Giant Tyres," has recently been issued by the Dunlop Rubber Co. Ltd., of Erdington, Birmingham. There are 32 pages in this booklet, which deals with many aspects of care and servicing of tyres for commercial vehicles. Among the features illustrated are: a gauge for checking camber, castor angle and king-pin inclination, a method of verifying that tyres are suitable for pairing, and a way of checking wheel alignment. The centre pages contain tables giving inflation pressures and permissible loads on tyres for rims of 16 in diameter and more, and for low-loading tyres for rims of 15 in diameter and less. Other tables are given for tyres for light commercial vehicles.

MEASURING INSTRUMENTS

New Apparatus for Production Control

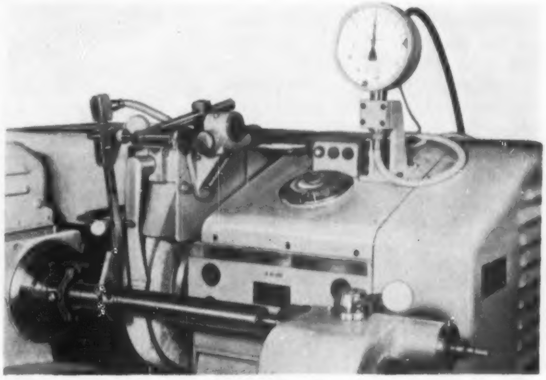
THREE new instruments have recently been introduced by C. E. Johansson Ltd. One is the C.E.J. Mikrokator, which is a mechanical comparator for indicating variations of component dimensions from those of a gauge block. Another is the C.E.J. Surface Finish Indicator; this operates on the tracer principle and records variations of surface profile. The third instrument is the C.E.J. Deltameter, which is a pneumatic unit for automatic control of grinding operations.

A noteworthy feature of the Mikrokator is that the amplifying mechanism between the measuring head and the indicator pointer is frictionless. This feature has been obtained by employing a twisted metal strip of rectangular cross section to actuate the pointer of the indicator dial. The strip is mounted horizontally in the instrument, and one half is twisted to the right and the other half to the left. Thus, when the strip is stretched, the centre portion to which the pointer is attached, rotates about the axis of the strip.

One end of the twisted strip is fixed to the body of the instrument and the other is attached to a strut on the end of a vertical spindle. The strut is set at a small angle from the horizontal in the same vertical plane as the strip so that, as the spindle is raised, this angle becomes smaller and the distance between the end of the strut attached to the strip and the fixed end of the strip increases. This causes the strip to untwist and move the pointer over the dial. The lower end of the spindle carries the measuring point, so that the movement of the pointer over the dial is an indication of the dimensions of the component being measured.

Checking the diameter of a limit plug gauge with a type 509, Mikrokator mounted on a stand

The least sensitive of these instruments can be used for measurement over a range of ±0·003 in and dial graduations are for every 0·0001 in, while the most sensitive have a range of ±0·0001 in and dial graduations for every 0·000002 in. At the back of each are provided two tolerance pointers, which are adjustable. The measuring point normally employed is a 5 mm diameter steel ball, which is extremely hard, but tips of Tungsten carbide or diamond can be supplied.



A cylinder grinding machine equipped with a type S, Deltameter for checking diameters

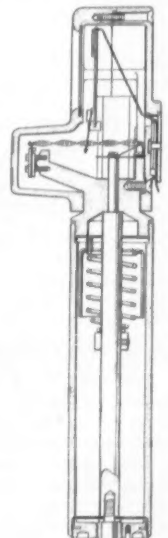
This instrument has several advantages. The movement of the spindle is amplified at the pointer without interference due to friction between solids and fluids, the response is instantaneous, there is no lost motion or wear and no drag due to congealed oil. Moreover, accurate measurements may be taken without time being wasted in stabilizing the equipment to suit varying temperatures.

The amplification device of the Surface Finish Indicator is based on the same principle as that of the Mikrokator. However, in this instrument, one end of the twisted strip is fixed but the other is actuated by the movement of the measuring needle, or stylus. Three, polished, cemented carbide skids under the base of the instrument support it on the surface. The stylus, which is a diamond tipped needle positioned at right angles to the base, is passed through a hole in one of the skids and pressed on to the surface with a force approximately equal to 1 gm. The apparatus can be traversed by hand or its motion can be accurately controlled by means of a micrometer feeder unit.

Surface Finish Indicators of different sensitivities are made. The most sensitive ones have a total range of ±0·0001 in and graduations for every 0·000002 in. Accessories are also available to adapt the unit for checking the finish of small machine parts and to support it in such a way that only the skid through which the stylus is passed rests on the test surface.

As has already been stated, the Deltameter is a pneumatically-operated instrument for the automatic control and measurement of grinding operations. It can be installed on, or built into, the grinding machine. The instrument can be supplied with air, through a moisture trap and pressure regulator, from the normal factory pipe line. It is a balanced measuring system based on pneumatic bellows so that air line pressure fluctuations do not in any way affect accuracy of the instrument.

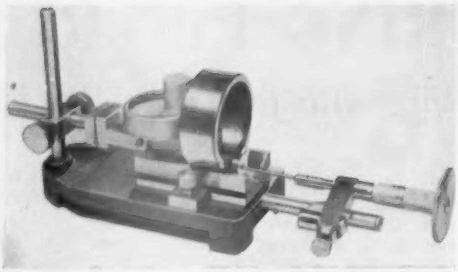
Movement of the measuring tip or stylus, when it is in contact with the workpiece, is transmitted to the Delta valve and thence to the dial indicator. This indicator is a separate component and may be mounted in any position on, or adjacent to, the machine. It is



A section through the Mikrokator

calibrated for two ranges of measurement, one for coarse and the other for fine grinding. The change over from one scale to the other is effected automatically.

After the roughing operation, the grinding wheel can be removed automatically and dressed, and fed anew into the workpiece; at the same time the fine division range on the dial will come into operation. The machine is automatically cut off when predetermined limits are reached and the Deltameter can operate an alarm signal if required. A range of heads to suit all purposes is manufactured. The machine can be used for the control of rolling mill strip or wire, and the machine can be automatically corrected or stopped when the thickness or width of the material exceeds a certain limit. Another adaptation is for the automatic sorting of the pieces into various groups. The advantages said to be obtained by using this instrument to control grinding or turning operations are: high precision, saving in time and fewer rejections. In most cases there is no need for inspection.



A Surface Finish Indicator being used in conjunction with a micrometer feeder

THE I.B.C.A.M. COMPETITIONS

Conditions and Awards for Drawing and Handicraft Sections

FOR many years a regular feature of the work of the Institute of British Carriage and Automobile Manufacturers has been to conduct Drawing and Handicraft competitions to stimulate interest in, and to extend the knowledge of, vehicle body design. Particulars of the 1954 series of competitions, arranged in conjunction with the Society of Motor Manufacturers and Traders, The Worshipful Company of Coachmakers and Coach Harness Makers of London, and the National Federation of Vehicle Trades, are now available.

Of the five drawing competitions, the first is for a long-distance touring saloon on a wheelbase of 10 ft 11 in. It is to be of advanced styling, have accommodation for four passengers in addition to a driver, provide amenities for Continental touring, and possess ample enclosed luggage space. Elevation and half-plan to a scale of 1½ in per ft and a coloured drawing ¾ in per ft are required.

A four-door, four-light, chassisless saloon, to be powered by an engine not exceeding 1·5-litre swept capacity, is the subject of the second competition. Dimensions and layout are entirely at the discretion of the competitor, but emphasis is given to suitability for large-scale production, and economy in tooling and assembly costs. A general arrangement drawing to a scale of 1½ in per ft is to be supplemented with full-size sections of details.

Restricted to students under the age of twenty-one, or other competitors under the age of twenty-three who have served not less than twelve months in H.M. Forces, competition No. 3 is for a two-door, two-seat, enclosed car of racing type on a wheelbase not exceeding 8 ft 3 in. The design must conform in details to the requirements stipulated in appendix C, part 2, of the General Competition Rules issued by the R.A.C.

Competitions No. 4 and No. 5 relate to passenger and commercial vehicles. No. 4 is for a full-fronted, luxury coach, of chassisless construction, suitable for use by a British operator. Overall dimensions are to be 30 ft by 8 ft and the design must comply in all respects with current Ministry of Transport regulations for single-deck vehicles.

Competition No. 5 is for a general arrangement of a 10 cwt general delivery van with forward control and a payload capacity not exceeding 150 ft^3. This, also, must be suitable for large-scale production and special consideration is to be given to practicability for K.D. shipment and overseas assembly.

Cash prizes are offered for the three most meritorious entries in each competition. The first prize in Competitions No. 2 and No. 4 is £75, in No. 1 and No. 5, £50, and in No. 3, £20. In addition, the first-prize winners in Competitions No. 1, No. 2, No. 4 and No. 5 will receive a supplementary award of hotel and travelling expenses (from London) for a visit to a Continental Motor Show. In adjudication, marks will be awarded on the basis of 25 per cent each for design, execution, practicability and originality.

The handicraft competition, open to junior entrants as in Competition No. 3, is in two sections, bodymaking and sheet metal work. Competitors are required to construct, respectively, a model of a front bulkhead frame for a single-deck bus body or an all-metal door for an open sports car, to drawings supplied by the Institute. A sum of £60 is allocated for awards at the discretion of the judges.

The Competitions are open only to persons of British nationality. Application for full conditions of entry and for the necessary uniformly sized drawing paper should be made to the Secretary, I.B.C.A.M., 50, Pall Mall, London, S.W.1, accompanied by a remittance of 2s. 6d. to defray costs. Entries must be returned on or before 22nd May, 1954.

CORRESPONDENCE

CLUTCH WITHDRAWAL BEARINGS

SIR,—We refer to the ball type thrust bearing manufactured by Pollard Bearings Limited and described in the December 1953 issue of *Automobile Engineer*, and would welcome your assistance in clarifying certain statements made as to the operation of clutch release bearings.

We feel that the references in the first paragraph of your article to the carbon block type of thrust bearing, and in the last paragraph to the standard clutch withdrawal bearing, imply that thrust rings, as fitted to all cars not using ball bearings, are made of plain carbon. It is true that carbon bearings, at times, were found to have a limited life, but all such bearings are now made of metallized carbon and experience has shown that their life is well above the normal major overhaul period of the car, that is, 45,000-50,000 miles.

All such bearings are of our manufacture and we would refer to the test described in the May 1951 issue of *Automobile Engineer*, page 183, where the total estimated life of 190,000 miles was obtained with metallized carbon release bearings, under worse than average conditions.

We question the possibility of substantiating the claim of a life of fifteen times such figures and submit that this relates to a bearing material that is no longer used, at any rate as first equipment.

THE MORGAN CRUCIBLE COMPANY LIMITED,
G. M. Slight,
Manager, Specialty Department.

THE BOEING GAS TURBINE

Application of a Gas Turbine Unit to Road Vehicles

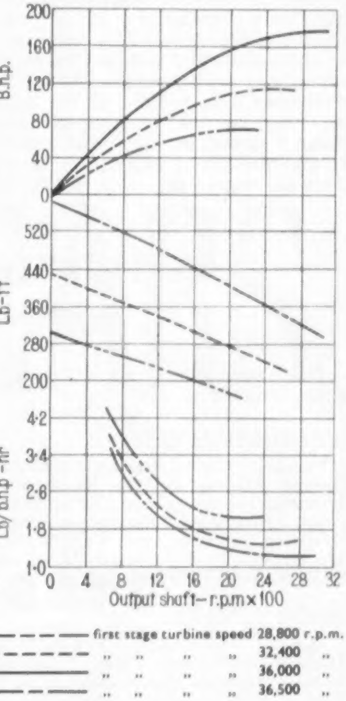
H. Wilkin Perry

IT seems that trucks, tractor-trailers and buses powered by gas turbines may, in the not too remote future, be available in the United States. This development has to a large extent arisen from pioneer work done by a major aircraft company in that country. A demonstration that supports this prospect was made in 1952, when a 55,000 lb heavy-duty Kenworth truck powered by a 240 lb gas turbine unit of 175 h.p. was driven 1,445 miles through the Pacific coast states from the Canadian border to the Mexican border, in less than 60 hours. This time included stops for refuelling and drivers' meals. The average speed of the vehicle while on the road was 33 m.p.h., and at no time were the state speed limits exceeded. After the test, the vehicle was displayed at the National Truck and Trailer Show in Los Angeles.

This turbine power unit was designed and built by the Boeing Airplane Co., not specifically for use in road vehicles, but under contract with the U.S. Navy. A unit of the same model and power has now been service-tested for more than a year in a truck operated, by West Coast Fast Freight Inc., on highways in Washington State in northwest U.S.A. adjoining Canada. It has hauled loads up to the legal limit of 68,000 lb gross weight over terrain that includes long steep gradients over the Cascade Mountain range. Units of the same design have been tested in a Navy personnel boat, a landing craft, an aeroplane and a helicopter, and others are being manufactured to supply electric power for Navy minesweepers.

The Boeing Company states that at present it has no plans for marketing this gas turbine for automobile use because two economic obstacles have yet to be overcome. One of these is the high cost, per unit, of manufacture on a small production basis. The other is the high rate of fuel consumption. However, in certain applications, specific advantages of the gas turbine outweigh these considerations and tend to make the unit suitable for heavy-duty commercial vehicles. These advantages are: small size, light weight, mechanical simplicity, ease of starting at all temperatures, favourable torque characteristics, relatively low cost of diesel fuel or paraffin, and reduction of maintenance time and cost. Nevertheless, these advantages fall far short of offsetting the disadvantages of high initial cost and high fuel consumption when the gas turbine is considered as a power source for private cars. The Company is striving constantly to reduce substantially the fuel consumption, and expects the manufacturing cost to decrease as the volume of production rises.

Model 502 is the latest version to be put into production. This unit weighs only 240 lb complete with the necessary auxiliaries, and its power output is comparable with a diesel truck engine weighing 3,000 lb or more. It has only one-tenth the number of parts and accessories as a petrol or diesel engine for the same application and needs no cooling system. The weight-saving with the gas turbine installed in a large commercial vehicle chassis of conventional design would enable more than one ton of additional payload per trip to be hauled. The overall dimensions of the unit are: length 40 in, width 23 in, and height 22 in. It occupies, therefore, approximately one-seventh of the space required for a diesel engine of equivalent power output.



Performance curves, Model 502

The latest Boeing gas turbine, Model 502, is now being manufactured for the U.S. Navy Bureau of Ships

Thus, if manufacturers should decide to introduce gas turbine-powered commercial vehicles, the full benefit could be obtained only by designing the chassis to take advantage of the small space required for the power unit by providing more room for the payload.

Such a development is logical and may come sooner than is generally expected. The General Motors Corporation has under way a research engineering project on a 300 h.p. gas turbine unit which may be used to power its inter-city motor coaches. Several other major manufacturers of trucks, buses, and automobiles are known to be carrying on similar studies, but are unwilling to talk about them. A prediction has been made by the director of the Ford Motor Co. scientific laboratory that turbine-type units will in time be lighter and cheaper than conventional power plants for both commercial vehicles and motor cars. Last August it was revealed that the Ford Motor

Co. has plans to build a research centre to test paraffin-burning turbine units for automobiles.

Now the Boeing Co. engineers are working on a plan for mounting two Model 502 turbines side-by-side in the same amount of space as that required for engines of trucks of conventional design. Their aim is at providing double the power for hill climbing and driving in heavy mud or sand. The two units are geared to a single propeller shaft. This arrangement will enable the driver to accelerate rapidly, using both the units when starting, and to operate on one turbine on good, level roads. The second unit may be brought into operation again for ascending steep gradients at a rate approximating to that of private cars, an accomplishment that will be much appreciated by motorists who now sometimes have to drive at ten miles per hour up hill for a considerable distance behind heavily loaded trucks. In mountainous or hilly terrain where there are many long, steep slopes, the doubled power will save much time. Moreover, heavier payloads can be carried because the twin turbine units weigh much less than a diesel engine of half the power.

Model 502 is described as a two-shaft unit composed of a compressor section and a power section with no mechanical interconnection. Therefore, the speed ratio is infinitely variable. The engine delivers 175 b.h.p. continuously at an output-shaft speed of 2,900 r.p.m.

A single-stage centrifugal air compressor at the front of the unit serves two stainless steel combustion chambers of tubular, straight-flow type. The flame tubes, or liners, are made of Inconel metal. Each flame tube has a fuel burner nozzle at its forward end and an ignition plug adjacent to the nozzle end. Either an electric or a compressed-air starter may be used. Air flow is at the rate of 3·6 lb/sec and the compression ratio is 3:1. Fuel is fed to the two burners by a single pump, which acts also as a governor. The maximum fuel pressure is 400 lb/in^2, and the consumption is 1·3 lb/b.h.p-hr.

The first-stage, axial-flow turbine wheel is 7 in diameter and has 64 blades. At rated power it operates at 1,550 deg F. The second-stage turbine is 9 in diameter and has 44 blades. It drives the output shaft through a planetary gear train with a reduction ratio of 8·63:1, and operates at rated power at a temperature of 1,350 deg F. Under these conditions, the exhaust temperature is 1,150-1,200 deg F. A sump of 4 qt. capacity is incorporated and a gear-type pump supplies the oil to the main bearings at a pressure of 80 lb/in^2. Because the unit has no reciprocating parts, there is little vibration when it is running, and except for the sound of the exhaust at high power output, it is almost silent. Power unit maintenance costs are low because there are so few running parts, no pistons and cylinders to wear, and no carburettor or high-pressure fuel injection pump, ignition apparatus, water pump or radiator.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during January:—

BIRMINGHAM CENTRE

Tuesday, 26th January, 6.45 p.m. General Meeting in the James Watt Memorial Institute, Great Charles Street. Paper: "Functions of Materials in Bearing Operation," by P. P. Love, B.Sc., Wh.Sc., M.I.Mech.E., P. G. Forrester, M.Sc., and A. E. Burke, B.A., G.I.Mech.E.

DERBY CENTRE

Monday, 25th January, 7.15 p.m. in the Midland Hotel, Derby. Paper: "Development of the Aston Martin Car," by R. L. Wyer.

NORTH-EASTERN CENTRE

Wednesday, 20th January, 7.30 p.m. General Meeting in the Chemistry Lecture Theatre, The University, Leeds. Paper: "The Jaguar Engine," by W. M. Heynes, M.I.Mech.E.

NORH-WESTERN CENTRE

Friday, 15th January, 7.15 p.m. General Meeting in Reynolds Hall, College of Technology, Manchester. Paper: "The Charging Processes of Internal Combustion Engines, with Special Reference to the Two-stroke Cycle," by Professor Dr. Hans List.

SCOTTISH CENTRE

Monday, 18th January, 7.30 p.m. General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "The Manufacture and Properties of Automobile Suspension Springs," by C. J. Dadswell, Ph.D., B.Sc., M.I.Mech.E., J. E. Russell, M.A., and R. Fielding.

WESTERN CENTRE

Thursday, 28th January, 6.45 p.m. General Meeting in the Grand Hotel, Bristol. Paper: "Functions of Materials in Bearing Operation," by P. P. Love, B.Sc., Wh.Sc., M.I.Mech.E., P. G. Forrester, M.Sc., and A. E. Burke, B.A., G.I.Mech.E.

The following meetings will be held during February:—

LONDON

Tuesday, 9th February, 5.30 p.m. General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper: "The Application of Power Assistance to the Steering of Wheeled Vehicles," by F. H. Heacock, A.M.I.Mech.E., and H. Jeffery, A.M.I.Mech.E.

COVENTRY CENTRE

Tuesday 2nd February, 7.15 p.m. General Meeting in the Craven Arms Hotel, Coventry. Paper: Address by the Chairman of the Centre, "Forty Years of Ignition and Lighting Development for Motor Cycles."

DERBY CENTRE

Monday, 8th February, 7.15 p.m. General Meeting in the Midland Hotel, Derby. Paper: "The Jaguar Engine," by W. M. Heynes, M.I.Mech.E.

LUTON CENTRE

Monday, 8th February, 7.30 p.m. General Meeting in the Town Hall Assembly Room. Paper: "Tyre Characteristics as Applicable to Vehicle Stability Problems," by T. J. P. Joy, B.Sc., A.M.I.Mech.E., and D. C. Hartley, B.A.

SCOTTISH CENTRE

Monday, 15th February, 7.30 p.m. General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Automobile Design in Retrospect." by J. A. Kemp, M.I.Mech.E.

FRICTION LININGS

A NEW booklet, entitled "Capasco Moulded Brake and Clutch Linings," has been published by the Cape Asbestos Co. Ltd., London, W.1. In its twelve pages, this publication gives details of the properties of Capasco linings. Curves showing the coefficient of friction plotted against brake drum temperature are given for a wide range of materials. The uses for which these materials are suited vary from small friction drives in window-winding mechanisms, for which an impregnated asbestos millboard-base material with a coefficient of friction of 0·28 is used, to a moulded brake lining with a coefficient of friction of 0·41, for touring cars. There are also heavy-duty, non-fade, moulded brake linings, which have a coefficient of friction of 0·36 up to a temperature of 620 deg C. Oil- and water-repellent features are also claimed for this type of lining. Another material, described as exceptionally resistant to fade, is for dry-plate and in-oil applications. The dry coefficient of friction of this material is 0·34 and when in-oil operation is called for, the design should be based on a coefficient of friction of 0·12.

MOSS

GEARS

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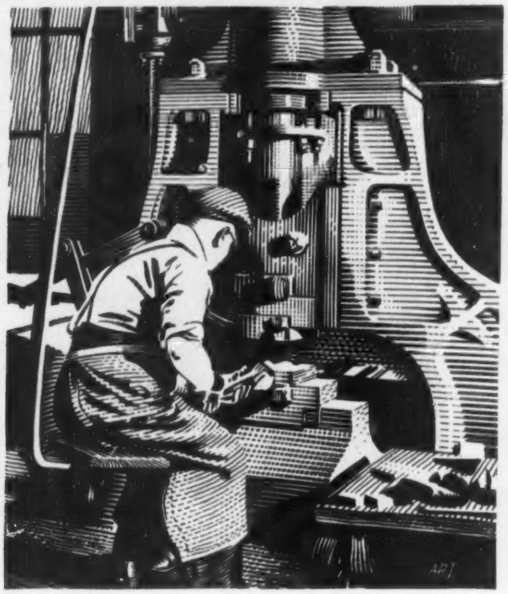

Beginning well . . .

Fifty years ago the engineer's choice of ferrous materials was simple—wrought iron, cast iron, mild steel, carbon steel. From these four he made his selection and built accordingly.

The present 'Age of Alloys' has made possible stresses, speeds and outputs hitherto unobtainable, and in this Doncasters of Sheffield were amongst the first to introduce and treat the alloy steels. These have since extended, and so revolutionised the whole field of steel usage.

From alloy steels and other metals of high quality, Doncasters shape forgings, drop forgings, tool steels, die blocks, hardened steel rolls and other products, coupling traditional craftsmanship and skills with the most modern of metallurgical methods and controls.

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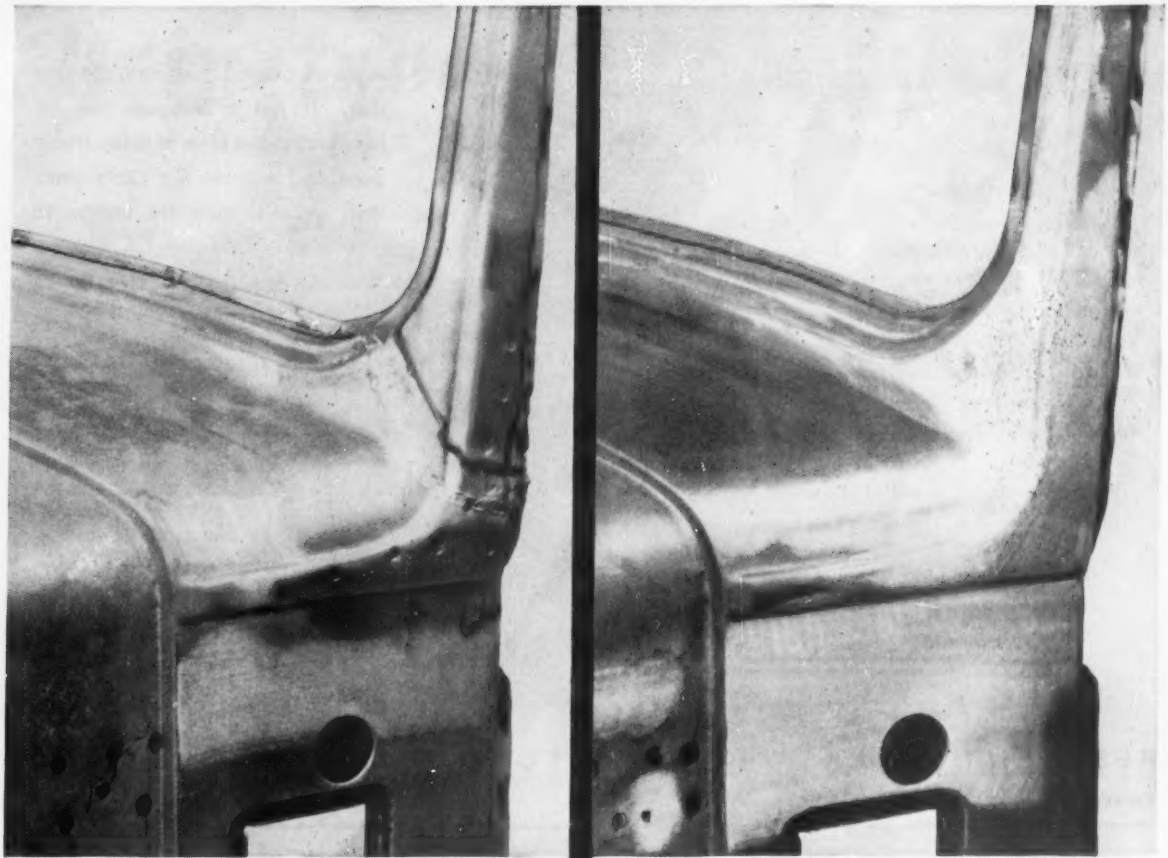
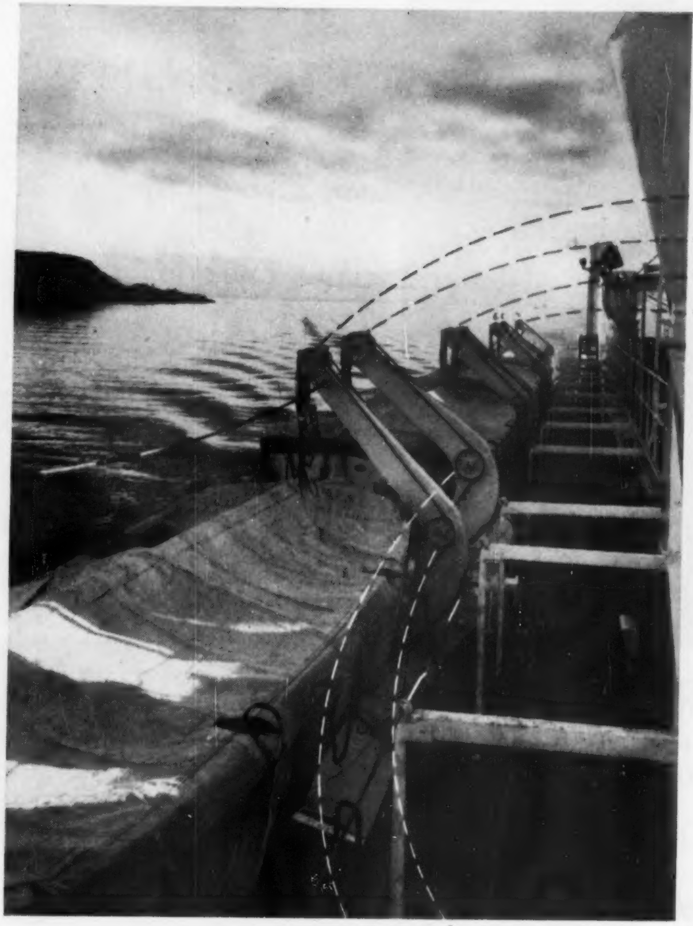
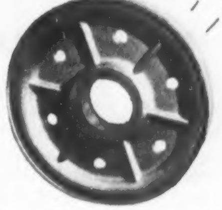


Photographs by courtesy of Pressed Steel Co. Ltd.



M.V. "Aureol" 14,083 g.t. built for the Elder Dempster Line.

MEEHANITE CASTINGS VERSUS THE CORRODING SEA



The sheaves and rollers of these davits made by Schat's Davits Ltd. for the Elder Dempster liner M.V. "Aureol", are cast in Meehanite metal. The choice of Meehanite metal was obvious—its anti-corrosion properties, close uniformity of grain and high strength were too important as safety factors to be ignored. With Meehanite the scientific balancing of materials and methods ensures that each casting has the precise properties required for the job. Whether you require small repetition castings or larger castings up to twenty tons, we can meet your need. Why not consult us on any of your present or future needs in castings?

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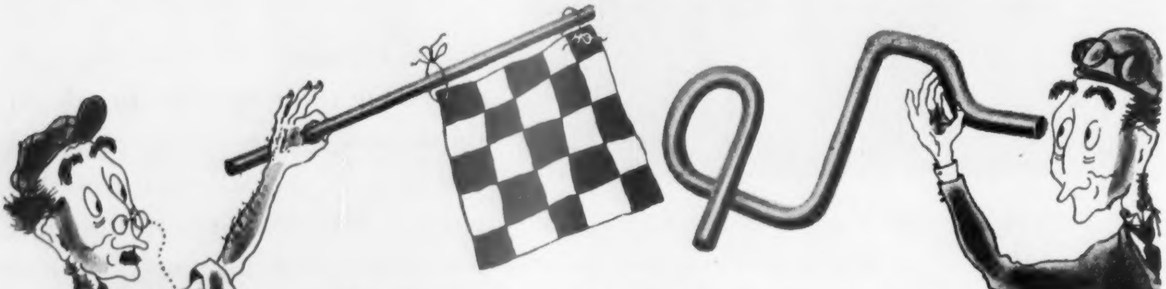


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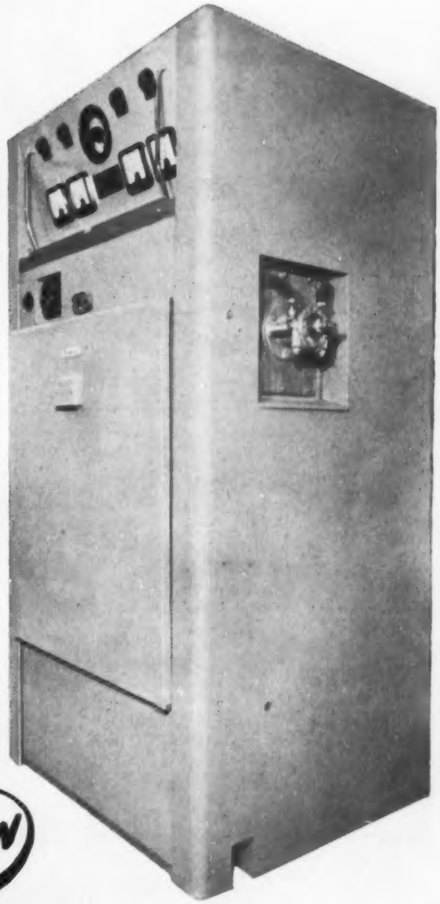

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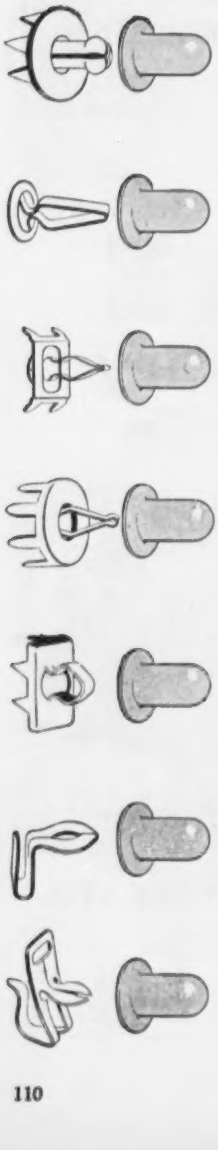

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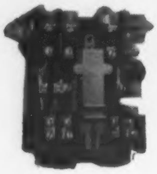
for single-phase or poly-phase non-reversing squirrel-cage induction motors.

This new starter is elegant in appearance, sturdy in construction, and dependable in operation. It is available for motors up to 5 horsepower, and complies with BS.587 for "Frequent Duty", i.e. forty starts per hour.



The starter, comprising a triple-pole contactor with normally-open auxiliary switch and triple-pole hand-reset thermal overload relay, is enclosed in a distinctive die-cast aluminium case with integral START-STOP push buttons.

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25-60 cycles

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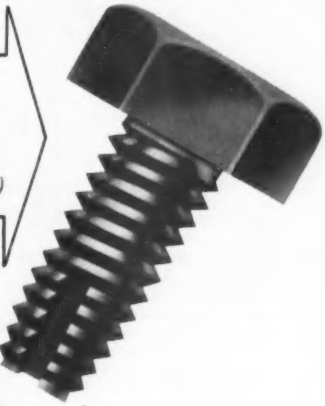

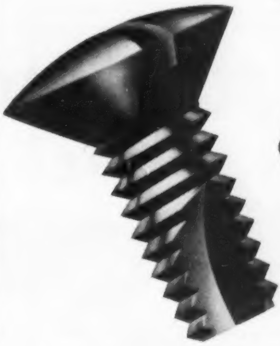

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PATENTS

THE proprietor of British Patent No. 627023, entitled "Improvements in or Relating to Vehicle Undercarriages" offers same for licence or otherwise to ensure practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, 14 E. Jackson Boulevard, Chicago 4, Illinois, U.S.A. [4755

SITUATIONS VACANT

The engagement of persons answering these advertisements must be made through the local office of the Ministry of Labour and National Service, etc., if the applicant is a man aged 18-64 or a woman aged 18-59 inclusive, unless he or she or the employer is excepted from the provisions of The Notification of Vacancies Order, 1952

DRAUGHTSMAN—Senior, with automobile experience for work on new project. Harry Ferguson Research Ltd., Fletchamstead Highway, Coventry. [4757

JOSEPH LUCAS Ltd. require a young Automobile Engineer for the coldroom section of their Proving Laboratory. Duties include investigations into the starting of I.C. engines under low temperature conditions and other allied problems. Applicants should have H.N.C. in mechanical engineering and experience in the testing of automobile engines is desirable. Please reply stating qualifications, experience and age to Personnel Manager, Joseph Lucas Ltd., Great King Street, Birmingham, quoting reference PM/D/88. [4758

MIDLAND firm requires capable mechanical engineering Draughtsman to work with chief engineer on project designs. The work, whilst covering a wide field, will generally be associated with all forms of transport.

ABILITY to do stress calculations essential. Good salary for right man. Box 2329, c/o Automobile Engineer. [4765

MECHANICAL Engineers for Product Design. Must have practical experience; University Degree desirable. Good prospects. 40-hour 5-day week. Pension and Life Assurance scheme. Write stating experience and salary required to International Harvester Company Limited, Doncaster. [4763

EXPERIMENTAL Engineers are required by F. Perkins Ltd., for research and development work on high speed diesel engines and their auxiliaries. Work and prospects are interesting. A pension scheme includes life insurance. Good technical qualifications are required, together with experience of experimental work (in industry) on high speed internal combustion engines. Age and brief details of experience must accompany applications to Personnel Manager, F. Perkins Ltd., Eastfield Factory, Peterborough. [4762

TECHNICAL Representative required for Midlands and North England for sales development of synthetic resins for foundry casting and polyester resins for glass laminates in engineering, automotive and aircraft fields. Some engineering and chemistry knowledge if possible. Give details of age and experience to Box 2510, c/o Automobile Engineer. [4767

SITUATIONS VACANT

MINISTRY of Transport and Civil Aviation: Certifying Officer. The Civil Service Commissioners invite applications for 2 pensionable posts. One in Birmingham and the other in Leeds or Manchester. Age at least 35 on 1st January, 1954. Candidates must be Corporate Members of the Institution of Mechanical Engineers and must be fully qualified mechanical and automobile engineers who have completed a recognized professional training.

INCLUSIVE remuneration £1,009 to £1,318. Somewhat lower for women. A starting salary above the minimum may be granted. Promotion prospects.

FURTHER particulars and application forms from Civil Service Commission, Scientific Branch, Trinidad House, Old Burlington Street, London, W.1, quoting S4272/54. Application forms to be returned by 20th January, 1954. [4764

DESIGNER-DRAUGHTSMEN are required by the Automobile Gearbox Division of the David Brown Gear Group of Companies.

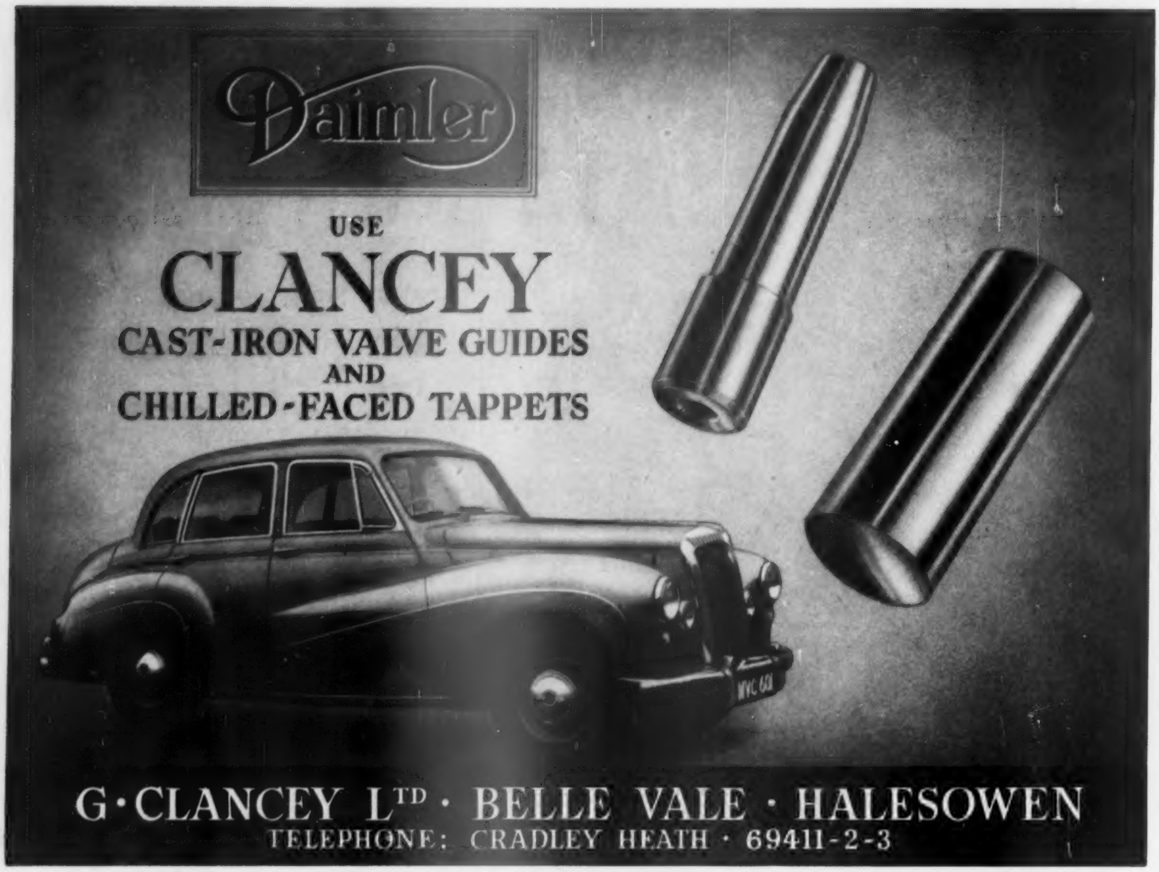
KNOWLEDGE of gears and their application in the Automobile Industry will be particularly useful.

ALTHOUGH there is no age limit, applicants are requested to mention their age in addition to full description of experience and salary requirements.

THESE will be monthly appointments, offering good salaries, Pension, Life Assurance and Sickness benefits and good working conditions.

MEN with technical training and sound experience are asked to write to the Chief Personnel Superintendent, David Brown & Sons (Huddersfield) Ltd., Park Gear Works, Huddersfield. [4766

RESEARCH and Development Worker required in the development department of large Midland motor firm. Higher National Certificate or equivalent. Write stating training experience and qualifications. Box 1667 c/o Automobile Engineer. [4751



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